Energy and Exergy Analysis of Ghazlan Power Plant

by

Jamil Jarallah Awadh Al-Bagawi

A Thesis Presented to the

FACULTY OF THE COLLEGE OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

October, 1994

ENERGY AND EXERGY ANALYSIS OF GHAZLAN POWER PLANT

BY

JAMIL JARALLAH AWADH AL-BAGAWI

A Thesis Presented to the

FACULTY OF THE COLLEGE OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

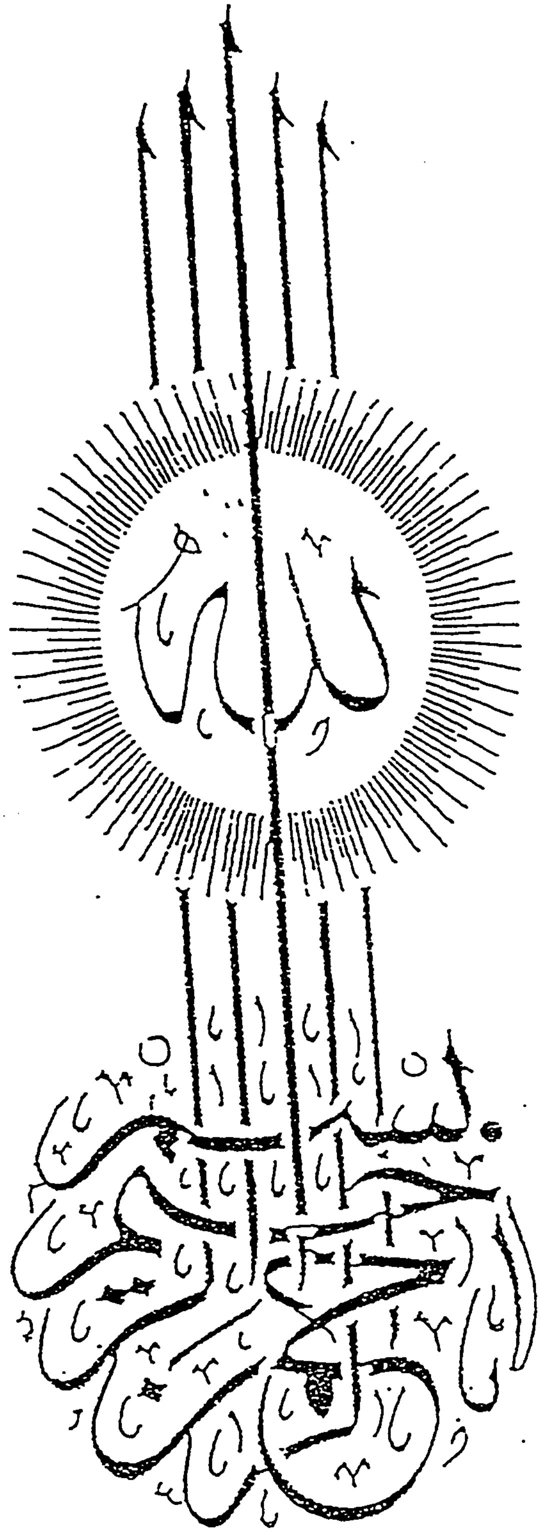
MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

OCTOBER 1994

UMI Number: 1361059



KING FAHD UNIVERITY OF PETROLEUM AND MINERALS

DHAHRAN 31261, SAUDI ARABIA

COLLEGE OF GRADUATE STUDIES

This thesis, written by Jamil Jarallah Awadh Al-Bagawi under the direction of his Thesis Committee, has been presented to and accepted by the Dean of the College of Graduate Studies, in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE in MECHANICAL ENGINEERING.

Thesis Committee

12/11/94

DR. SYED AHMED M. SAID
Thesis Advisor

Med Habib, 12/11/94

DR. MOHAMED ABDEL-AZIZ HABIB
Co-Advisor

12/11/94

DR. MAGED AHMED EL-SHAARAWI
Member

Nov. 9, 1994

DR. SYED M. ZUBAIR
Member

DR. MOHAMMED O. BUDAIR
Department Chairman

DR. ALA AL-RABEH
Dean of Graduate Studies

14.11.94

Date

COLLEGE OF GRADUATE STUDIES

ACKNOWLEDGEMENT

Thanks be to ALLAH ALMIGHTY for all the knowledge he has given to me. Also, praise and gratitude be to ALLAH ALMIGHTY, with whose gracious help it was possible to accomplish this work.

I gratefully acknowledge the Mechanical Engineering Department of King Fahd University of Petroleum and Minerals (KFUPM) for the support provided to carry out this research.

I wish to express my warmest and sincere gratitude to Dr. Syed Ahmed Said, my thesis committee chairman and Dr. Mohammed Habib my thesis Co-Advisor for their positive guidance, advice and relentless encouragement. I am also grateful to other members of my thesis committee, Dr. Majed El-Shaarawi and Dr. Sayed Zubair, for their significant contributions and valuable suggestions. Last, but certainly not least, I am very thankful to my beloved mother, father, wife, daughter, brothers, sisters and friends for their pray, patience, encouragement, support and understanding.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Figure No.		Page No
4.19	Effect of Reheat Steam Temperature on the Performance	62
4.20	Effect of Reheat # 1 and # 2 Pressures on the First Law Efficiency	64
4.21	Effect of Reheat # 1 and # 2 Pressures on the Total Irreversibility	65
4.22	Effect of Rehat # 1 and # 2 Pressures on the Second Law Efficiency	66
4.23	Effect of Number of Reheat on the Performance and Fuel Flow Rate	67
4.24	Dimensionless Plot of Improvement in Heat Rate due to Feed Heating [32]	69
4.25	Effect of Extracted Pressures from Turbines to Feedwater Heaters on the First Law Efficiency	71
4.26	Effect of Extracted Pressures from Turbines to Feedwater Heaters on the Second Law Efficiency	72
4.27	Cycle Arrangement for Six Feedwater Heaters	73
4.28	Effect of Number of Feedwater Heaters on the Performance	74

THESIS ABSTRACT

FULL NAME OF STUDENT : **JAMIL JARALLAH A. AL-BAGAWI**

TITLE OF STUDY : **ENERGY AND EXERGY ANALYSIS OF GHAZLAN POWER PLANT**

MAJOR FIELD : **MECHANICAL ENGINEERING**

DATE OF DEGREE : **OCTOBER, 1994**

The design and actual thermodynamic performance of Ghazlan power plant have been studied based on both first and second -law analysis. A comparison between the two performances indicated that there is a room for improvements. Being encouraged by this and the fact that a 1% improvement in performance will result in saving of millions of Saudi Riyals per year, a full exergy analysis is carried out to identify the potentials for improving the plant performance. The exergy analysis shows a detailed breakdown of exergy losses of the different components in the plant. Hence based on the results of the analysis, several alternative arrangements to improve the plant efficiency were considered. This included the plant turbine cycle operation under various conditions. In this regard, the thermodynamic physical quantities whose effects were studied are: throttle steam pressure and temperature, reheat steam pressure and temperature for a single reheat, double reheat, reheat pressure for a double reheat and number of feedwater heaters. The results of the study are presented in tabular and graphical forms and discussed in detail.

MASTER OF SCIENCE DEGREE

KING FAHD UNIVERSITY OF PETROLEUM AND MINERALS,

Dhahran, Saudi Arabia

October 1994

ملخص الرسالة

اسم الطالب : جميل جارالله عوض البقعاوي

عنوان الرسالة : التحليل الطاقي والأكسرجي (exergy) لمحطة غزلان الحرارية

التخصص : الهندسة الميكانيكية

تاريخ الرسالة : اكتوبر ١٩٩٤م.

تمت دراسة الأداء الديناميكي الحراري التصميمي والحقيقي لمحطة غزلان الحرارية بناء على تحليل القانون الأول والقانون الثاني الديناميكي الحراري . اظهرت مقارنة هذه النتائج أن هناك إمكانية لعمل تحسينات في هذه المحطة. وبتشجيع من هذه النتائج وحقيقة ان رفع كفاءة المحطة ١ ٪ يعني توفير ملايين الريالات السعودية سنويا ، وقد تم عمل تحليل أكسرجي (exergy) كامل لتحديد إمكانية تحسين كفاءة هذه المحطة . حيث بين التحليل الأكسرجي الخسائر في مكونات المحطة. وبناءعلى ذلك ، تمت دراسة عدة خيارات لتحسين كفاءة المحطة . و تتضمن دراسة دائرة توربينة المحطة التي تعمل تحت عدة خيارات . القيم الفيزيائيه التي تم دراسة تأثيرها هي : ضغط بخار الخانق ، درجة حرارة بخار الخانق ، ضغط بخار معيد التسخين الاحادي ،درجة حرارة بخار معيد التسخين الاحادي، معيدالتسخين الثنائي ، ضغط بخار معيد التسخين الثنائي وعدد مسخنات الماء المغذي . وهذه النتائج تم عرضها بجداول ورسوم توضيحية ونوقشت بالتفصيل.

درجة الماجستير في العلوم
جامعة الملك فهد للبترول والمعادن
الظهران - المملكة العربية السعودية
اكتوبر ١٩٩٤ م

CHAPTER 1

INTRODUCTION

1.1 LITERATURE REVIEW

Attempts to improve the performance of power plants are a never ending subject. Until recently all of these attempts were based on the first law analysis and limited by economic and/or technical considerations. Recently, the adequacy of the methodology was challenged by the use of the exergy concept based on the second law of thermodynamics. Many of the prominent thermodynamicsts [1, 2] have pointed the utilization of the exergy concept as based on the second law of thermodynamics. Utilization of the exergy concept to analyze steam power plants results in accurate evaluation of the available energy dissipations and hence more meaningful results as compared to those obtained using the first law. Results show that the effects of some parameters are significantly different for the first and second law analysis. Applications of this important but not intensively used concept of exergy in power plant analysis have been reported in a number of articles in the literature.

El-Masri [3] presented a thermodynamic methodology based on the second-law for analyzing gas turbine combined cycles. El-Masri [4] used the exergy analysis concept to calculate and provide a detailed breakdown of the sources of inefficiency of a combined cycle. In this regard, he developed a computer program known as GASCAN for the thermodynamics analysis of the cycles. He showed that the dominant interaction governing the variation of cycle efficiency with turbine inlet temperature is that between combustion irreversibility and turbine cooling losses. Also a detailed analysis and loss breakdown of steam bottoming cycle based on the exergy concept was presented by El-Masri [5]. El-Masri [6] developed a computer code which performs exergy balance analysis to break down and trace system inefficiencies to their source components and source processes within the components of a gas turbine system. Porz [7] developed a computer method for thermodynamic power cycle calculations to model any cycle scheme by selecting components from a library. Tsatsaronis and Sanae [8] developed a computer software for on-line thermoeconomic evaluation of steam power plant. They discussed the preliminary results obtained from the software. Mastrullo [9] presented a software package designed to accomplish a detailed thermodynamic analysis of Rankine cycle and to provide the thermodynamic properties in a fixed state. Spakovsky and Evans [10] carried out performance optimization of thermal systems in which operation costs are based on the second law. They presented an accurate picture of where the true cycle inefficiencies exist. Habib [11] carried out a comparison between a cogeneration and a conventional plant based on the first and second laws analysis. The distribution of irreversibility rates for the different components of the conventional and gas-turbine co-generation plants at the same process mass and power output were presented. Wong, et al. [12] optimized a power cycle at different loads by minimization of irreversibilities. They found out that when the power cycle operates at full load, the main steam temperature and pressure should be at the upper limit for minimal irreversibilities in the system and it should be decreased for a load factor of less than 65%. Bidini and Stecco [13] proposed a method (TEXAS) that derives families of characteristic curves showing economic parameters versus exergetic efficiency for industrial plants. Bollant [14] presented a comparison of measures to improve the efficiency of combined gas and steam turbine cycles. His calculations were based on the first and second law approaches which provide a good tool for the analysis of power cycle.

Habib and Zubair [15] examined the performance of a regenerative Rankine cycle power plant based on the exergy concept. The distributions of irreversible losses in various components of the cycle were presented. Habib and Zubair [16] also examined the performance of regenerative-reheat power plants in terms of irreversibility analysis. The reduction in the irreversible losses with the addition of backward, cascade type feedwater heaters and/or a reheat option were compared with a conventional energy-balance approach. Gaggioli, et al. [17] presented exergy flow diagrams for a combined power and desalination facility in Abu Dhabi. The exergy diagrams show the rates of exergy flow between all the components in the facility as well as the consumption in each. Sciubba and Su [18] presented the results of an extensive first and second law analysis of the performance of the turbine cycle for steam power plants. The results show a striking differences between the two analyses, in terms of the prediction of the influence of the various parameters on the performance, both in a qualitative and a quantitative sense.

1.2 PLANT CHARACTERISTICS AND CONFIGURATION

The characteristics of the employed power plant are those of Ghazlan power plant which consists of four units. The design power output of each unit is 400 MW. The capacity of the plant is 1600 MW from four units. Each turbine is a double-flow exhaust and operated at 3600 rpm. Turbine inlet steam conditions are 1800 psi, 510°C. There are four boilers, each boiler is capable of producing 2,850,000 pounds of steam per hour (1,293,000 Kg/h). Heat rate average is 11,500 BTU/kWh. The primary fuel is blended natural gas. Fuel oil, crude oil, or naphtha are used as backup fuels (primarily Arabian light crude oil is used also as backup fuel). The stack is made of concrete with an acid resistant cement lining and is about 91 meters high. Arabian Gulf water is used as a coolant in condenser. After filtration to remove the debris and marine life, the clean water is circulated through the condenser. Two pumps, each rated at 137,000 gallons per minute, are required for each unit. The water is returned to the Arabian Gulf. A desalination plant with a capacity of 2,500,000 gallons per day, provides the necessary desalted water for the plant. A major portion of water is supplied to the nearby Fractionation Center. Two tanks are provided for storing the condensate.

Electricity is generated at 24,000 volts. Voltage is stepped up to 230,000 volts by transformers. Power is delivered through transmission lines to the interconnected grid networks for further delivery to industrial complexes, crude oil production facilities, natural gas liquefaction centers, and the residential communities.

The power plant configuration showing the major components of the power plant is shown in Fig. 1.1. It consists of (1) the steam generator (furnace + heat exchangers), and (2) steam turbine cycle (turbines + heaters + condenser + pumps).

A brief description of the plant operation is as follows: fuel is burned in the furnace where much of the heat generated is transferred to boiler tubes which line the walls. The steam-water mixture formed in the boiler tubes proceeds to a steam drum, where the vapor and liquid are separated. The liquid is returned to the boiler tubes and the vapor proceeds to a super-heater. There the hot gases leaving the flame region super-heat the saturated steam produced in the boiler section. The super-heated steam proceeds to the turbine, where its thermal energy is converted to mechanical energy. The steam leaving the high-pressure stage of the turbine is reheated in the furnace prior to being sent to the intermediate pressure sections and then to the low-pressure sections of the turbine. The steam exiting from the low-pressure turbine proceeds to the condenser which will produce a vacuum or

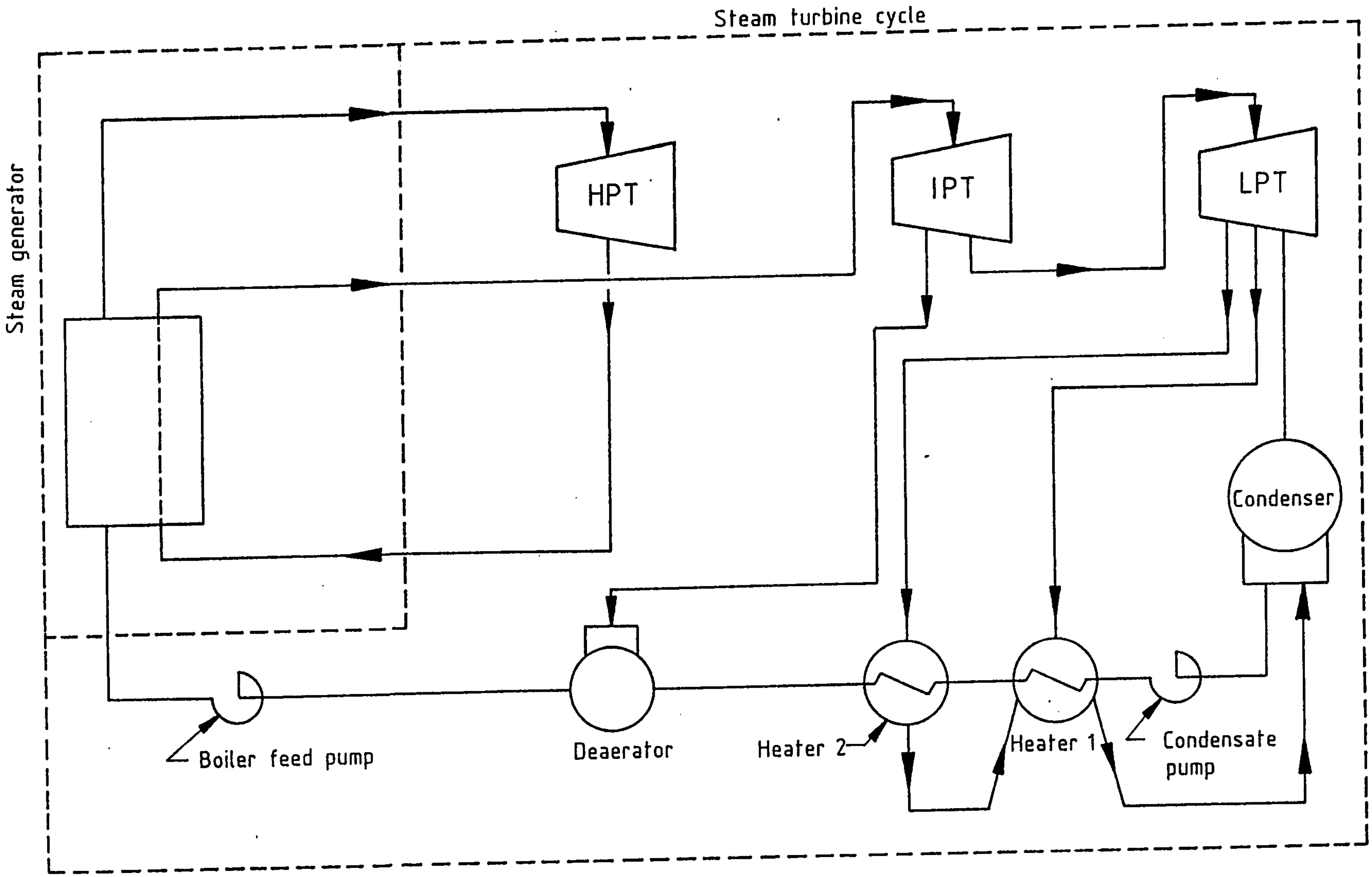


Fig. 1.1 Power plant configuration.

desired back pressure at the turbine exhaust. The liquid from the condenser is reheated in two regenerative feed water heaters, using steam bled from the turbine. The feedwater then proceeds to the deaerator, which is an open feedwater heater, to remove dissolved gases, especially oxygen and CO_2, from the boiler feedwater, thereby reducing corrosion levels throughout the system. The feedwater then proceeds to the boiler feed pump which will increase the boiler pressure. The feedwater exiting the boiler feed pump will proceed to the furnace, where it is brought to near saturation in the economizer. The gases leaving the economizer preheat the entering air and then discharged through the stack.

1.3 OBJECTIVE

The objective of this study is to carry out an extensive energy (first law) and exergy (second law) analysis of the performance of an existing 1600 MW (fuel-fired) electrical power plant to identify the potential for improvement. The design and actual conditions will be studied. Also, the effect of throttle steam pressure and temperature, reheat steam pressure and temperature, and effect of number of feedwater heaters, both in a qualitative and quantitative sense will be discussed. The results of analyses using both the first and second law of thermodynamics for the turbine cycle of the plant operating under various conditions will be presented.

CHAPTER 2

GOVERNING THERMODYNAMIC EQUATIONS

A power cycle is mathematically represented by a system of algebraic equations. The system of equations which comprise the mathematical model employed consists of (1) the mass, energy, and exergy balance equations, (2) the thermochemical property relations, and (3) the associated boundary conditions.

1. The Balance Equations:

For steady-state steady-flow process, the conservation of mass is

$$\sum_e \dot{m}_e = \sum_i \dot{m}_i \tag{1}$$

The conservation of energy is

$$\sum_i \dot{E} + \dot{Q}_{c.v} = \sum_e \dot{E} + \dot{W}_{c.v} \tag{2}$$

The exergy balance is

$$\sum_i \dot{A} + \sum_j \left(1 - \frac{T_o}{T_j}\right)\dot{Q}_{c.v} = \sum_e \dot{A} + \dot{W}_{c.v} + \dot{I} \tag{3}$$

The chemical species balance is

$$\sum_e \dot{N}_j = \sum_i \dot{N}_j + \dot{N}_p \tag{4}$$

2. The Thermochemical Property Relations:

Neglecting variations in potential and kinetic energies, the convective energy rate of stream i is

$$\dot{E}_i = \sum_j \dot{N}_{ij} \bar{h}_j \tag{5}$$

while the total flow exergy of stream i is

$$\dot{A}_i = \dot{N}_i \bar{a}_{ti} \tag{6}$$

where the total specific flow exergy a_{ti} of a given flow stream i is

$$\bar{a}_{ti} = \sum_j X_{ij} \bar{a}_t \tag{7}$$

and the specific flow exergy a_t is

$$\bar{a}_t = \bar{a}_{TM} + \bar{a}_{CH} \tag{8}$$

where :

$$\bar{a}_{TM} = (h - h_o) - T_o(S - S_o) \tag{9}$$

and

$$\bar{a}_{CH} = \bar{R} T_o \ln \frac{Y_i}{Y_i^e} \tag{10}$$

Energy Evaluation

For a steady state process, and neglecting the kinetic energy and potential energy, the conservation of energy equation (2) can be written as

$$\dot{Q}_{c.v} + \sum_i \dot{m}_i h_i = \dot{W}_{c.v} + \sum_e \dot{m}_e h_e \tag{11}$$

Applying the conservation of energy equation on each component of the plant, the following expressions will be obtained.

Steam Generator (Boiler) :

(a) <u>The Heat Exchanger</u> :

$$W_{c.v} = 0$$

$$\dot{Q}_{c.v} = \dot{Q}_B$$

$$\dot{Q}_B = \sum_e \dot{m}_e h_e - \sum_i \dot{m}_i h_i \tag{12}$$

(b) <u>Furnace (Combustion Chamber)</u> :

The available energy of reactants is given by:

$$\dot{Q}_{in} = \dot{m}_f * HHV \tag{13}$$

where :

$\dot{m}_f$ is the mass flow rate, and

HHV is the fuel higher heating value.

The steam generator thermal efficiency is defined as

$$\eta_b = \frac{\dot{Q}_B}{\dot{Q}_{in}} \tag{14}$$

Turbines:

Assuming the turbine is adiabatic $(\dot{Q}_{c.v} = 0)$,

$$\dot{W}_{c.v} = \dot{W}_t$$

$$\therefore \qquad \dot{W}_t = \sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e \qquad (15)$$

Condenser :

For the condenser :

$\dot{W}_{c.v} = 0$

$\dot{Q}_{c.v} = -\dot{Q}_c$ since the heat is rejected.

$$\therefore \qquad \dot{Q}_c = \sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e \qquad (16)$$

Pumps :

Assuming the pump is adiabatic $(\dot{Q}_{c.v} = 0)$,

$\dot{W}_{c.v} = -\dot{W}_p$ since the work is done on the working fluid.

$$\therefore \qquad \dot{W}_p = \dot{m}_e h_e - \dot{m}_i h_i \qquad (17)$$

Feedwater Heaters :

For the feedwater heaters

$$\dot{W}_{c.v} = 0$$

Taking the control volume on the water side

$$\dot{Q}_{F.H} = \left[\sum_e \dot{m}_e h_e - \sum_i \dot{m}_i h_i\right]_{\text{Water side}} \tag{18}$$

Taking the control volume on the steam side

$$\dot{Q}_{F.H} = \left[\sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e\right]_{\text{Steam side}} \tag{19}$$

Deaerator :

It is an open feedwater heater. Assuming no heat losses to the surrounding

$$\therefore \quad \dot{Q}_{c.v} = 0$$

and

$$\dot{W}_{c.v} = 0$$

$$\therefore \quad \sum_e \dot{m}_e h_e = \sum_i \dot{m}_i h_i \tag{20}$$

Electric Generator :

The output power produced by the generator is given by

$$P = \eta_g * \dot{W}_t \tag{21}$$

where η_g is the efficiency of the generator.

The first law thermal efficiency of the overall power plant is given by:

$$\eta_I = \frac{P}{\dot{m}_f * \text{HHV}} \tag{22}$$

or

$$\eta_I = \frac{P}{\dot{Q}_{in}} \tag{23}$$

Exergy Evaluation

Expressing the second law principle in the same format as the conservation of mass and energy principles yields the availability (exergy) rate balance equation. Equation (3) can be rearranged and written as

$$\dot{I}_{c.v} = \sum_j \left(1 - \frac{T_o}{T_j}\right)\dot{Q}_{c.v} - \dot{W}_{c.v} + \sum_i \dot{m}_i a_i - \sum_e \dot{m}_e a_e \tag{24}$$

where **a** is the specific flow availability defined as

$$a_j = (h_j - h_o) - T_o (S_j - S_o) \tag{25}$$

<u>**Exergetic Efficiency**</u> **:**

In general, the second law efficiency is defined as the ratio of exergy desired (output) to the exergy needed (input). For the turbine cycle unit, the exergy output is the power, while the exergy input is the rate at which availability (exergy) is carried out at any combination of reheat pressures.

Hence, the second law efficiency of the turbine cycle unit is:

$$\eta_{II_t} = \frac{P}{\sum_i \dot{m}_i a_i - \sum_e \dot{m}_e a_e} \tag{26}$$

For the steam generator unit, the exergy output is the exergy input to the turbine cycle unit while the exergy input is

$$Q_{in} = \dot{m}_f * HHV$$

Hence the second law efficiency of the steam generator unit is

$$\eta_{II_G} = \frac{\sum_e \dot{m}_e a_e - \sum_i \dot{m}_i a_i}{\dot{Q}_{in}} \tag{27}$$

For the overall power plant, the exergy output is the power output while the exergy input is Q_{in}. Hence, the overall power plant second law efficiency is

$$\eta_{II_o} = \frac{P}{\dot{m}_f * HHV} \tag{28}$$

which is equal to the overall plant first law efficiency given by equation (22).

Irreversibility Evaluation

With the conceptual basis that entropy is that particular extensive property of matter, the natural entropy measure of the collective internal irreversibilities is termed as irreversibility (exergy destruction). The irreversibility can be determined from the availability rate or alternatively by evaluating the entropy production from an entropy rate balance and multiplying by T_o. The arguments in favor of selecting an entropy measure of irreversibility rather than an availability measure are favored.

Hence, in general, an entropy rate balance for a control volume and its surounding reduces at steady state to

$$\dot{\sigma} = \left[\left(\sum_e \dot{m}_e S_e - \sum_i \dot{m}_i S_i\right) - \frac{\dot{Q}_{c.v}}{T_j}\right] \tag{29}$$

where $\dot{\sigma}$ is the entropy production rate and the term Q_j/T_j represents the time rate of entropy transfer at the location on the boundary where the instantaneous temperature is T_j.

The irreversibility rate is given as

$$\dot{I} = T_o \dot{\sigma} \tag{30}$$

where T_o is the dead-state temperature.

Based on the above, the irreversibility rates of the plant two units components are given below:

Turbine Cycle Unit :

Using equations (29) and (30), the irreversibility rates of the turbine cycle unit can be obtained as shown below.

<u>Turbine</u> :

The irreversibility rate for the turbine is given as:

$$\dot{I}_t = T_o\left[\sum_e \dot{m}_e S_e - \sum_i \dot{m}_i S_i\right]_t \tag{31}$$

The turbine second-law efficiency is defined as:

$$\eta_t = \frac{\dot{W}_t}{\dot{W}_{rev}} \tag{32}$$

where :

$$\dot{W}_t = \sum_i \dot{m}_i h_i - \sum_e \dot{m}_e h_e \tag{33}$$

and

$$\dot{W}_{rev} = \sum_i \dot{m}_i (h_i - T_o S_i) - \sum_e \dot{m}_e (h_e - T_o S_e) \tag{34}$$

Using equations (31), (33), and (34) it can be readily seen that

$$\dot{W}_{rev} = \dot{W}_t + \dot{I}_t$$

Hence, the turbine second law efficiency can be written as:

$$\eta_t = \frac{\dot{W}_t}{\dot{W}_t + \dot{I}_t} \tag{35}$$

Condenser :

The condenser irreversibility rate is given as:

$$\dot{I}_{cond} = T_o\left(\left[\sum_e \dot{m}_e S_e - \sum_i \dot{m}_i S_i\right]_{cond} - \frac{\dot{Q}_c}{T_o}\right) \tag{36}$$

Feedwater Heater :

The feedwater heater irreversibility rate is given as:

$$\dot{I}_{F.H} = T_o\left[\sum_e \dot{m}_e S_e - \sum_i \dot{m}_i S_i\right]_{F.H} \tag{37}$$

Pump:

The pump irreversibility rate is given as:

$$\dot{I}_P = T_o\left[\dot{m}_e S_e - \dot{m}_i S_i\right]_P \tag{38}$$

Generator :

The generator irreversibility rate is given as:

$$\dot{I}_G = \dot{W}_t - P \tag{39}$$

while the generator efficiency is given as :

$$\therefore \quad \eta_G = \frac{P}{\dot{W}_t}$$

Steam Generator Unit (Boiler) :

Using equations (29) and (30) the irreversibility rates of the steam generator unit components can be obtained using the following expressions.

Heat Exchanger :

The irreversibility rate for the heat exchanger is given as:

$$\dot{I}_b = T_o\left[\left(\sum_e \dot{m}_e S_e - \sum_i \dot{m}_i S_i\right) - \frac{\dot{Q}_b}{T}\right] \tag{40}$$

Furnace :

Neglecting dissociation at elevated temperature, the chemical reaction equation can be expressed as : [21]

$$\sum_{l=1}^{L}\left[f_l C_{n_l} H_{m_l} + \alpha f_l\left(n_l + \frac{m_l}{4}\right)O_2 + 3.718\ \alpha f_l\left(n_l + \frac{m_l}{4}\right)N_2\right] + cCO_2 + dN_2 \rightarrow$$

$$(\Sigma f_l n_l + c)\, CO_2 + \Sigma \frac{m_l f_l}{2} H_2O + (\alpha - 1)\left[\Sigma f_l\left(n_l + \frac{m_l}{4}\right)\right] O_2$$

$$+\left(\left[3.76\ \alpha f_l\ \Sigma\left(n_l + \frac{m_l}{4}\right)\right] + d\right) N_2 \tag{41}$$

where

f_l is the molar percentage of $C_{nl}\ H_{ml}$ per one mole of fuel

c is the molar percentage of CO_2 per one mole of fuel

d is the mole percentage of N_2 per one mole of fuel

$(\alpha - 1)$ is the percentage excess air by volume

Defining X_i as the number of moles of component i in the product per unit mole of fuel, the following expression can be obtained:

$$X_{CO_2} = \Sigma f_l n_l + c \tag{42}$$

$$X_{H_2O} = \Sigma \frac{m_l f_l}{2} \tag{43}$$

$$X_{O_2} = (\alpha - 1)\left[\Sigma f_l \left(n_l + \frac{m_l}{4}\right)\right] \tag{44}$$

$$X_{N_2} = 3.76\,\alpha \left[\Sigma f_l \left(n_l + \frac{m_l}{4}\right)\right] + d \tag{45}$$

Hence the total number of moles of the product per unit mole of fuel will be given as

$$X_{tot} = \Sigma X_i = X_{CO_2} + X_{H_2O} + X_{O_2} + X_{N_2} \tag{46}$$

defining Y_i as the mole fraction of component i in the product, the following relations can be obtained:

$$Y_{CO_2} = \frac{X_{CO_2}}{X_{tot}}$$

$$Y_{H_2O} = \frac{X_{H_2O}}{X_{tot}}$$

$$Y_{O_2} = \frac{X_{O_2}}{X_{tot}}$$

$$Y_{N_2} = \frac{X_{N_2}}{X_{tot}}$$

The availability destruction in the flue gases can be presented by the sum of the thermo-mechanical availability and the chemical availability of the stack gases as given by equation (8).

The thermo-mechanical contribution in the availability destruction in the stack gases can be expressed as

$$\dot{A}_{TM}\left(\frac{kJ}{kmole\ of\ fuel}\right) = \Sigma X_i\left[(\bar{h}-\bar{h}_o) - T_o(\bar{S}-\bar{S}_o)\right]_i \tag{47}$$

Substituting equations (42-45) into equation (47), the following is obtained:

$$\therefore\ \dot{A}_{TM} = (\Sigma f_l n_l + c)\left[(\bar{h}-\bar{h}_o) - T_o(\bar{S}-\bar{S}_o)\right]_{CO_2}$$

$$+ \Sigma\frac{m_l f_l}{2}\left[(\bar{h}-\bar{h}_o) - T_o(\bar{S}-\bar{S}_o)\right]_{H_2O}$$

$$+(\alpha-1)\left[\Sigma f_l\left(n_l + \frac{m_l}{4}\right)\right]\left[(\bar{h}-\bar{h}_o) - T_o(\bar{S}-\bar{S}_o)\right]_{O_2}$$

$$+\left(3.76\,\alpha\left[\Sigma f_l\left(n_l + \frac{m_l}{4}\right)\right] + d\right)\left[(\bar{h}-\bar{h}_o) - T_o(\bar{S}-\bar{S}_o)\right]_{N_2} \tag{48}$$

The term $(\bar{S}-\bar{S}_o)$ can be written as

$$(\bar{S}-\bar{S}_o) = \bar{S}(T) - \bar{S}(T_o) - \bar{R}\ln(Y_i) \tag{49}$$

$$\dot{A}_{TM}(kW) = \frac{\dot{m}_f}{M}\Sigma X_i\left[(\bar{h}-\bar{h}_o) - T_o(\bar{S}-\bar{S}_o)\right]_i \tag{50}$$

where the molecular weight of fuel (M) is given as:

$$M\left(\frac{kg}{kmole}\right) = \sum_{l=1}^{L} f_l * M_{C_{n_l}H_{m_l}} + c * M_{CO_2} + d * M_{N_2} \tag{51}$$

The chemical contribution in the availability destruction in the flue gases can be expressed as:

$$\dot{A}_{CH}\left(\frac{kJ}{\text{kmole of fuel}}\right) = \bar{R}T_o \, \Sigma X_i \ln\left(\frac{Y_i}{Y_i^e}\right) \tag{52}$$

and

$$\dot{A}_{CH}(kW) = \frac{\dot{m}_f}{MW}\bar{R}T_o \, \Sigma X_i \ln\left(\frac{Y_i}{Y_i^e}\right) \tag{53}$$

where Y_i^e is the mole fraction of CO_2, H_2O, O_2, or N_2 in the environment and are given the following values:

$$Y^e_{N_2} = 0.7567; \; Y^e_{O_2} = 0.2035; \; Y^e_{H_2O} = 0.0303; \; Y^e_{CO_2} = 0.0003$$

Substituting equations (42-45) and the values Y_i^e into equation (52), the following is obtained:

$$\dot{A}_{ch} = \bar{R}T_o \, [X_{CO_2} \ln\left(\frac{Y_{CO_2}}{0.0003}\right) + X_{H_2O} \ln\left(\frac{Y_{H_2O}}{0.0303}\right)$$

$$+ X_{O_2} \ln\left(\frac{Y_{O_2}}{0.2035}\right) + X_{N_2} \ln\left(\frac{Y_{N_2}}{0.7567}\right)] \tag{54}$$

The irreversibility rate in the furnace is given as

$$\dot{I}_{c.c}(kW) = \dot{A}_f - \dot{A}_{TM} - \dot{A}_{CH} - \dot{A}_Q \tag{55}$$

Where the furnace availability input ($\dot{A}_f$) and output ($\dot{A}_Q$) are given as:

$$\dot{A}_f = \dot{m}_f \; * \; HHV = \dot{Q}_{in}$$

and

$$\dot{A}_Q = \dot{Q}_{in}\left(1 - \frac{T_o}{T_j}\right)$$

and $\dot{A}_{TM}$ and $\dot{A}_{CH}$ are given by equations (50) and (53).

CHAPTER 3

NUMERICAL PROCEDURE

A power cycle is mathematically represented by a system of algebraic equations. With the computer-aided power cycle calculation, numerous program packages have been developed to solve the resulting equations. The complexity of these programs covers a wide range. It reaches from programs that can be used for a few fixed cycle schemes, up to packages that allow the modelling of any cycle configuration. Packages that are restricted to a few cycle schemes usually lack the flexibility that is required for detailed investigations. On the other hand highly flexible all-purpose packages require detailed information about the cycle's topology, parameters, calculation sequence and convergence method. This makes them very difficult to handle. Hence, it was decided to develop a computer program that will handle the case under study efficiently.

The development goal was a computer program for studying conventional power cycles and in particular to be flexible, easy to use and expandable.

It is intended to use the program for the first- and second-law analysis of Ghazlan power plant by solving the algebraic equations being described in the mathematical formulation chapter. The sequential solution method will be used, in which it is possible to carry out the calculations sequentially unit after unit following the direction of mass flow. New unit models can be added easily and limited memory is required with this method. Solutions are carried out under design and actual operating conditions.

3.1 DESCRIPTION OF THE PROGRAM

The program consists of the main program and many subroutines. In addition, the subroutines prepared to cover the steam tables are included [29]. The main program has the following sections:

1. Nomenclature.
2. Input Data.
3. Calculation of enthalpy and entropy at each location of the plant.
4. Calculation of heat input to the plant, boiler, heat rejected in the condenser, heat exchanged in the feedwater heater.
5. Calculation of turbine work and generator output.
6. Calculation of pump work.
7. Calculation of irreversibility for each component in the plant.
8. Calculation of first law efficiency and second law efficiency.
9. Computer output printing.

The program document is shown in Appendix A and a sample of results is shown in Appendix B.

The flow chart of the developed computer program is shown in Fig. 3.1. The steps shown are exactly the same as the computer program takes. The input variables of the program are, fuel composition, fuel flow rate, steam flow rate, pressure and temperature at each location, and the steam leakage in each component. The output variables of the program are; the heat input for the plant and boiler, the heat rejected in the condenser, the heat exchanged in the feedwater heater, the work of turbines, the generator output, the work of the pumps, the irreversibility of each component of the plant, and the first and second law efficiencies.

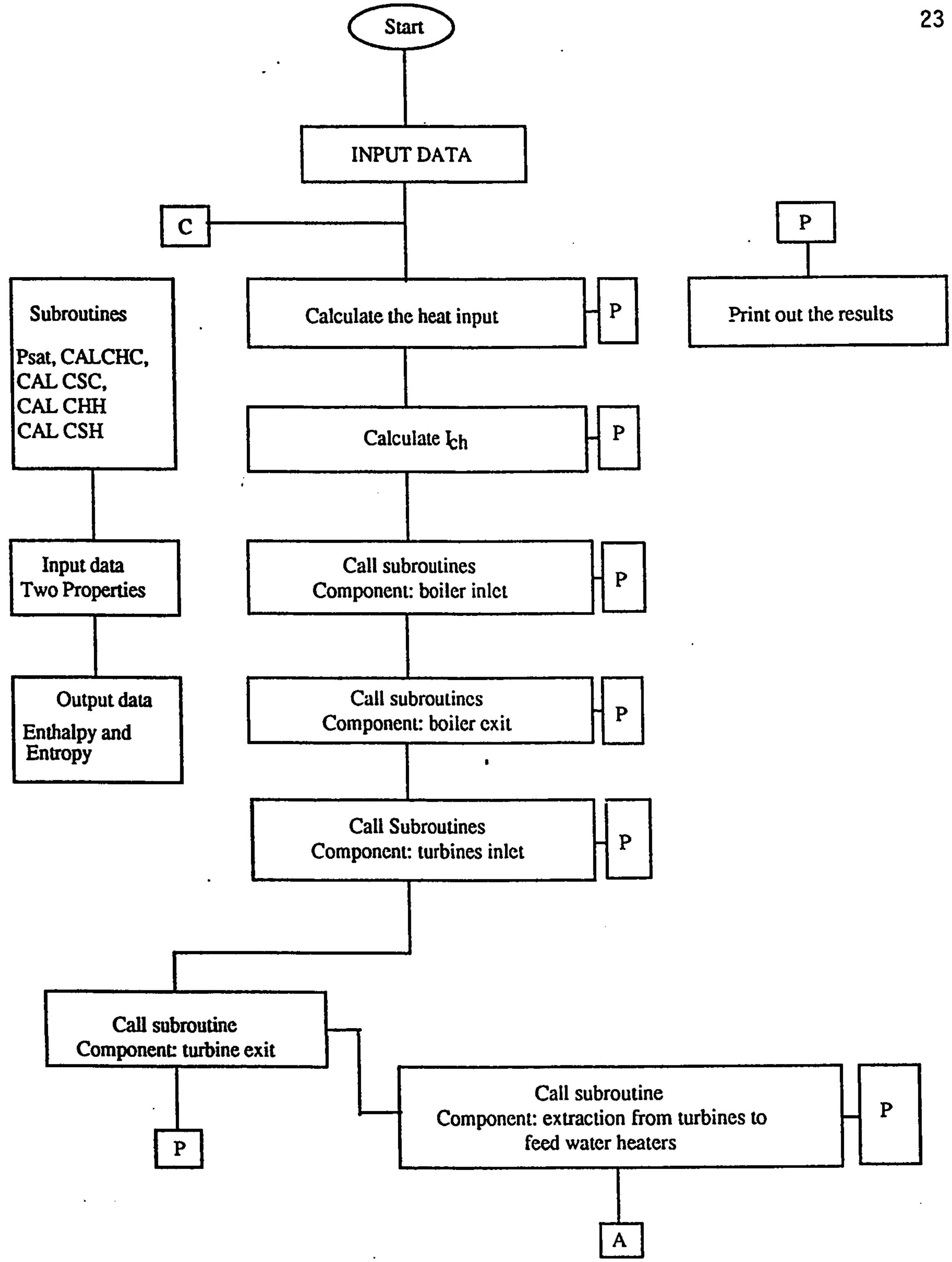
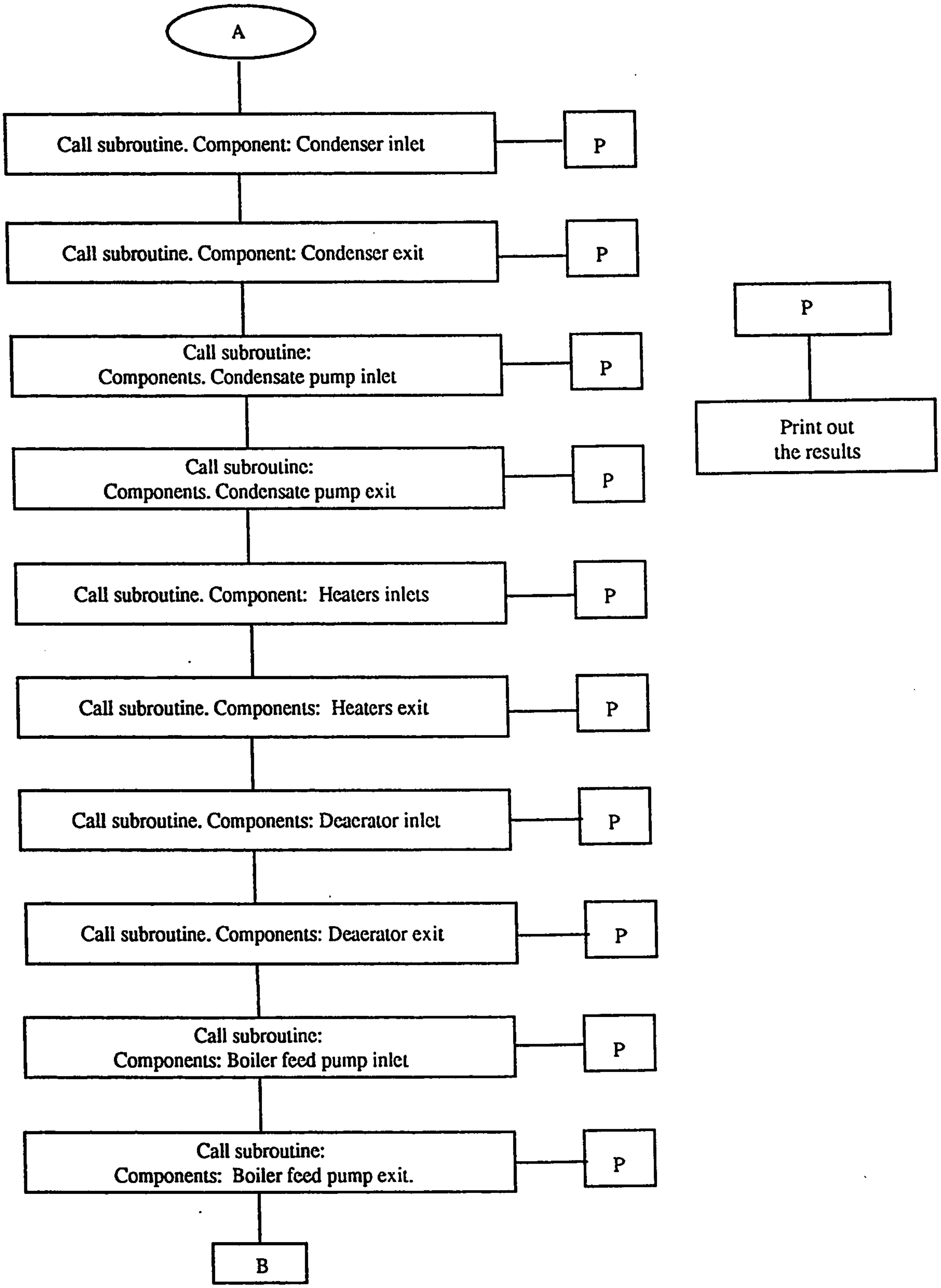
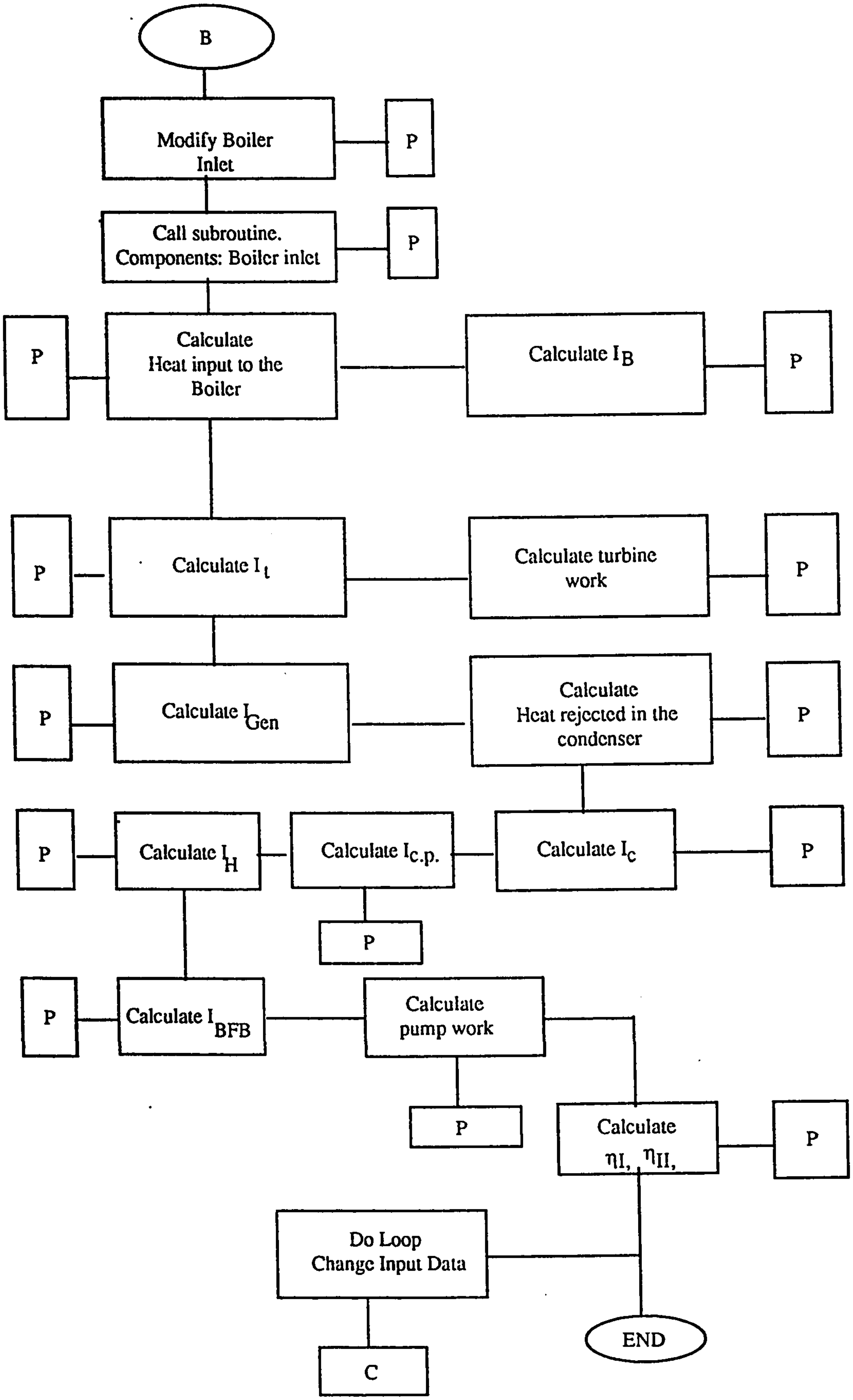


Fig. 3.1. Flow Chart of the Program



Continued Figure 3.1



Continued Figure 3.1

3.2 CAPABILITY OF THE PROGRAM

With minor changes the program can be used to study the effects of variation over a broad range of operating conditions and physical quantities, such as

1. throttle steam pressure,
2. throttle steam temperature,
3. reheat pressure,
4. reheat temperature,
5. number of reheat,
6. extracted pressure from turbines to feedwater heaters,
7. number of feedwater heaters
8. percent load, and
9. design alternatives.

3.3 THE VALIDITY OF THE PROGRAM

To validate the program and check its accuracy the following comparative studies were carried out:

(i) The thermodynamic state properties obtained using the thermodynamic properties subroutine were compared with the ones obtained using thermodynamic steam tables. The results shown in Table 3.1 exhibit an excellent agreement with a percentage difference of less than 0.01.

(ii) Using the program, the power and the efficiency were calculated for the plant design conditions at full load and compared with those values obtained from the design sheets. The results shown in Table 3.2 exhibit an excellent agreement with a maximum percentage difference of less than 1.

The comparative results shown in Tables 3.1 and 3.2 verified that the program was properly emplemented and hence can be used to carry out the necessary calculations.

3.4 BOUNDARY CONDITIONS

The boundary conditions for the power plant are: (i) the incoming air and fuel temperature is 25 ^{o}C; (ii) the flue gas temperature is 200 ^{o}C; (iii) all components, except the steam generator have adiabatic boundaries; and (iv) 15% excess combustion air.

Input data	Results using		Comparison
	Steam tables h(kJ/kg) s(kj/kg.k)	Program h(kJ/kg) s(kj/kg.k)	% difference
T=500 C P=3.0 Kpa	h=3456.5 s=7.2338	h=3456.18 s=7.23448	0.009 % 0.009 %
T=400 C P=150 Kpa	h=3277.4 s=8.3555	h=3277.45 s=8.35619	0.002 % 0.008 %
Saturated steam at 95 °C	h=2668.1	h=2667.94	0.006 %
Saturated water at 95 °C	h=397.96	h=397.93	0.008 %
Saturated steam at 140 °C	s=6.9299	s=6.9305	0.009 %
Saturated water at 140 °C	s=1.7391	s=1.7392	0.006 %

Table 3.1 Comparative Study (i)

	Results using		Comparison
	Design sheet	Program	% difference
Output Power (MW)	384.495	386.6	0.286%
Efficiency	35.45 %	35.77 %	0.902 %

Table 3.2 Comparative Study (ii) at Full Load.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 PLANT PERFORMANCE

Engineers are always curious to know how much better the performance of a plant could have been. Hence, they will thus need not just a performance parameter, which is solely a measure of performance, but a performance criterion against which the measured value of first- and second- law efficiencies can be compared. The design performance parameters will be used as a performance criterion in this study. The characteristics of the employed power cycle are those of an existing 1600 MW (fuel-fired) electrical power plant. The plant configuration is shown in Fig. 1.1 and the heat balance diagram at full load is shown in Fig. 4.1. The thermodynamic state properties of the power cycle are fixed at the values shown in Table 4.1. Since there is no flame temperature measurement, the actual average gas temperature is assumed to be 1600°K [31]. Increasing the flame temperature from 1300 to 2000°K will increase the steam generator heat exchanger irreversibility rate by 8% and will decrease the furnace irreversibility rate by an equivalent amount as illustrated in Fig. 4.2.

The first set of results to be presented pertains to the plant design performance. The results are presented in Figs. 4.3 through 4.7. Figure 4.3 presents the first law efficiency (η) and the total irreversibility (I) (as a percent of the input exergy) versus the load. The figure indicates an increase in η and decrease in I with an increase in the load percent. The rate of change in η and I decreases as the load increases. The load is normally reduced by reducing the fuel flow rate. Figure 4.4 shows the fuel flow rate ratio versus the load. As

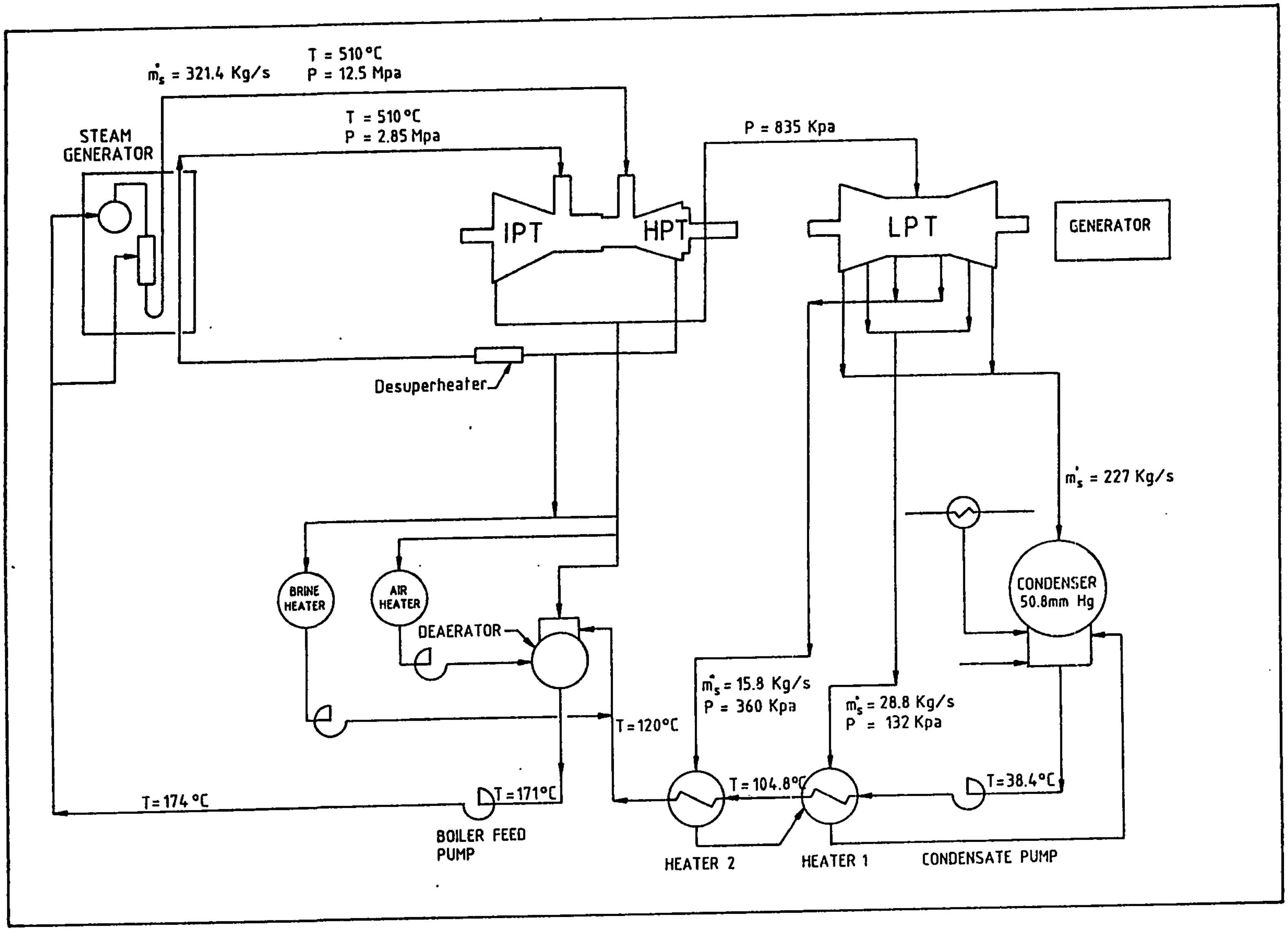


Fig. 4.1 Heat balance diagram.

Table 4.1. Values of the Thermodynamic Properties of the Power Cycle. [31]

	Actual Data	Design Data
Fuel Flow Rate, kg/s	1) 15.89 2)18.55 3) 19.89 4) 20.16	1) 6.67 2) 11.46 3) 16.05 4) 21.35
Steam Produced, kg/s	1) 205.55 2) 225.6 3) 252.8 4) 253.3	1) 93.05 2) 165.51 3) 237.30 4) 321.40
Flame Temperature*, ^oK	1600	1600
Fuel Gas Temperature*, ^oK	450	450
Excess Air*	15 %	15 %
Feedwater Temperature to boiler, ^oC	1) 153.33 2) 165 3) 169.44 4) 170.55	1)129.70 2) 146.80 3) 160.70 4) 174.10
Throttle Pressure, MPa	1) 12.418 2) 12.411 3) 12.404 4) 12.39	1) 12.514 2) 12.514 3) 12.514 4) 12.514
Throttle Temperature ^oC	1) 509.44 2) 507.78 3) 510.00 4) 511.11	1) 477.11 2) 504.56 3) 510.00 4) 510.00
Reheat Pressure, kPa	1) 1840.0 2) 2185.7 3) 2496.0 4) 2385.0	1) 716.40 2) 1401.8 3) 2045.1 4) 2850.4
Reheat Temperature, ^oC	1) 510.00 2) 510.00 3) 510.00 4) 511.11	1) 457.22 2) 488.40 3) 510.00 4) 510.00

Cont'd. on next page

Continued Table 4.1.

Exhaust Pressure, mm.Hg	1) 42.16 2) 53.84 3) 72.39 4) 65.27	1) 50.8 2) 50.8 3) 50.8 4) 50.8
Fuel Composition on a molar basis	89.4% CH_4 8.6% C_2H_6 0.4% C_3H_8 0.1% N_2 0.06% CO_2	89.4% CH_4 8.6% C_2H_6 0.4% C_3H_8 0.1% N_2 0.06% CO_2

* Assumed

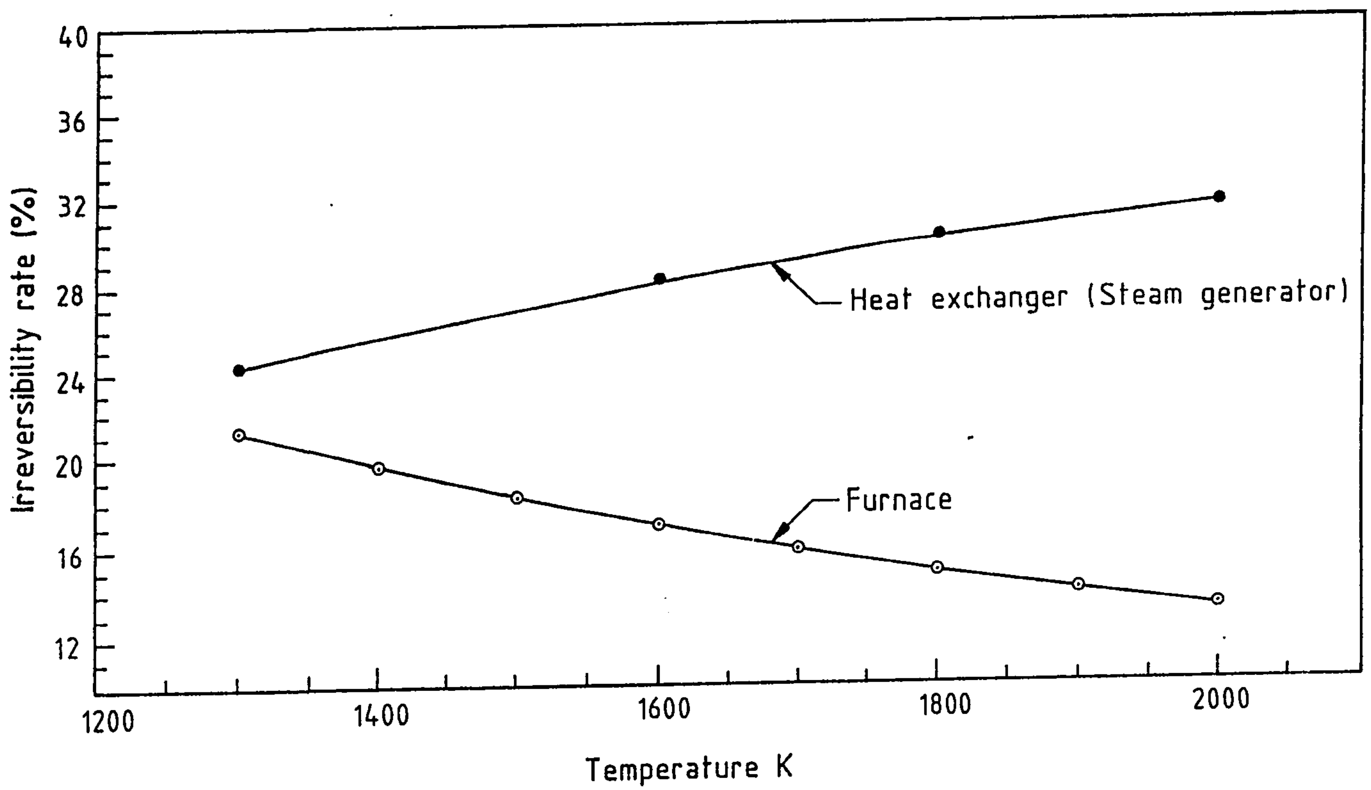


Fig. 4.2 Effect of the flame temperature.

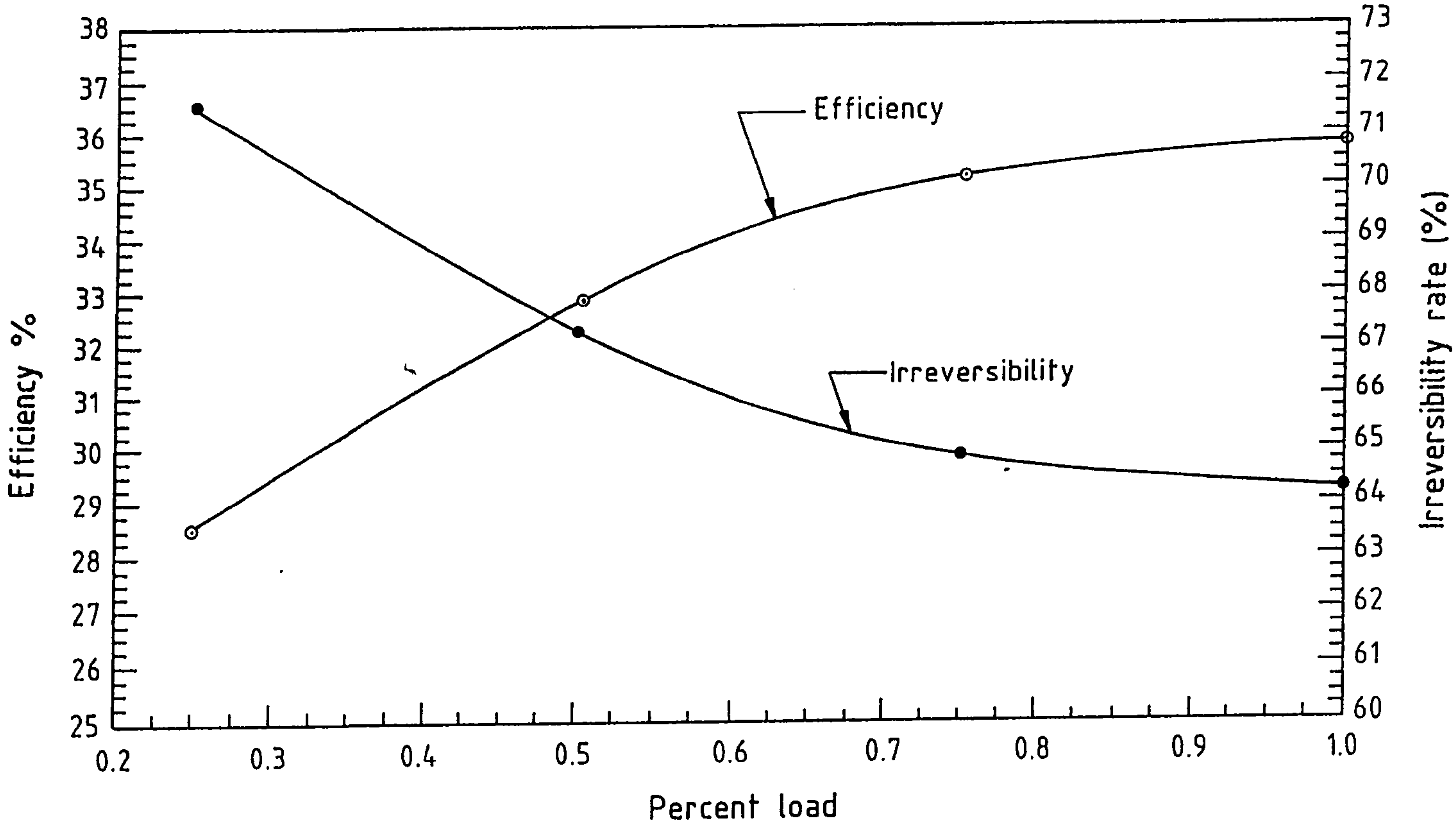


Fig. 4.3 Efficiency and irreversibility versus percentage of full capacity.

can be seen, the reduction in first law efficiency (η) at low loads can not be explained through Fig. 4.4.

The reduction in η at low loads may be attributed to irreversibility losses in the steam generator unit and the turbine cycle unit. As the load increases, the steam generator efficiency is expected to improve and the percent heat loss of the energy input in the steam generator is expected to decrease. This is confirmed by the results shown in Fig. 4.5 which indicate that the irreversibility loss decreases as the load increases. As can be seen, the irreversibility at full load is less than that at 1/4 load by 7.5%. Irreversibility losses in the steam generator are due to furnace irreversibility and heat transfer irreversibility. These losses are related to the thermomechanical and chemical irreversibilities which are expressed by equation (8). The losses are characterized by the gas losses and the difference in temperature between the boiler gases and the water-steam flow.

The h-s expansion lines for the high, intermediate, and low pressure turbines are shown in Fig. 4.6. The figure indicates clearly the decay of the efficiency of the high pressure turbine as the load is reduced. The 1/4 load h-s curve for the high pressure turbine exhibits a different trend compared to the others due to the fact that the decay in efficiency is also partly due to throttling that is normally considered at such low loads. The figure, thus explains the improvement in η at higher loads. Figure 4.7 shows the results of turbine irreversibilities as a function of load. These results reveal that the reduction in η of the turbine is due to irreversibilities of the throttling process. The figure indicates a reduction of irreversibility loss by 2% as the load increases from 1/4 to full load. Figure 4.8 contains the results of the overall plant and component exergetic losses as a percentage of input exergy versus the load. The enlargement of Figure 4.8 is shown in Fig. 4.9 for irreversibility rate between 45% to 73% of the input energy.

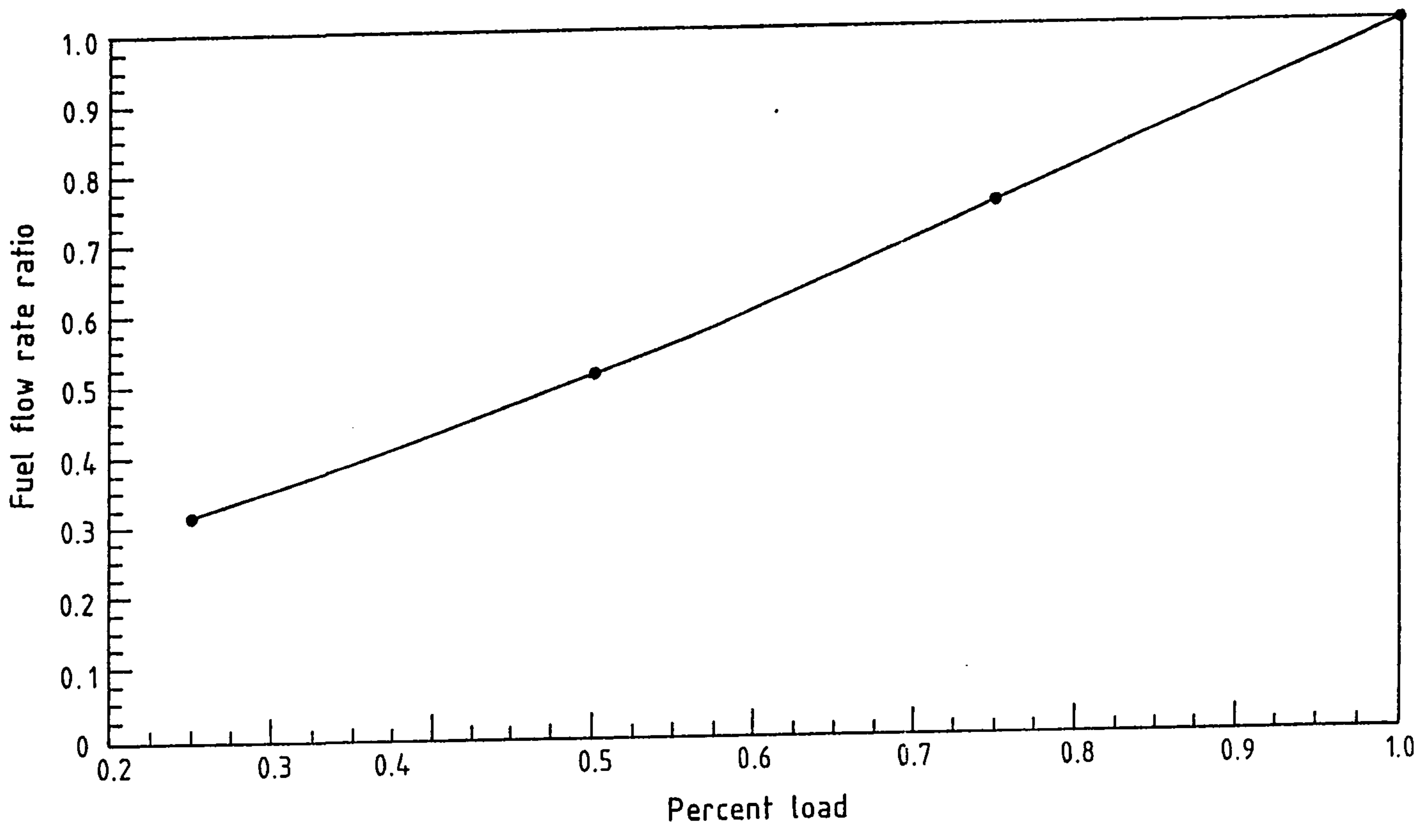


Fig. 4.4 Fuel flow rate ratio versus percent load

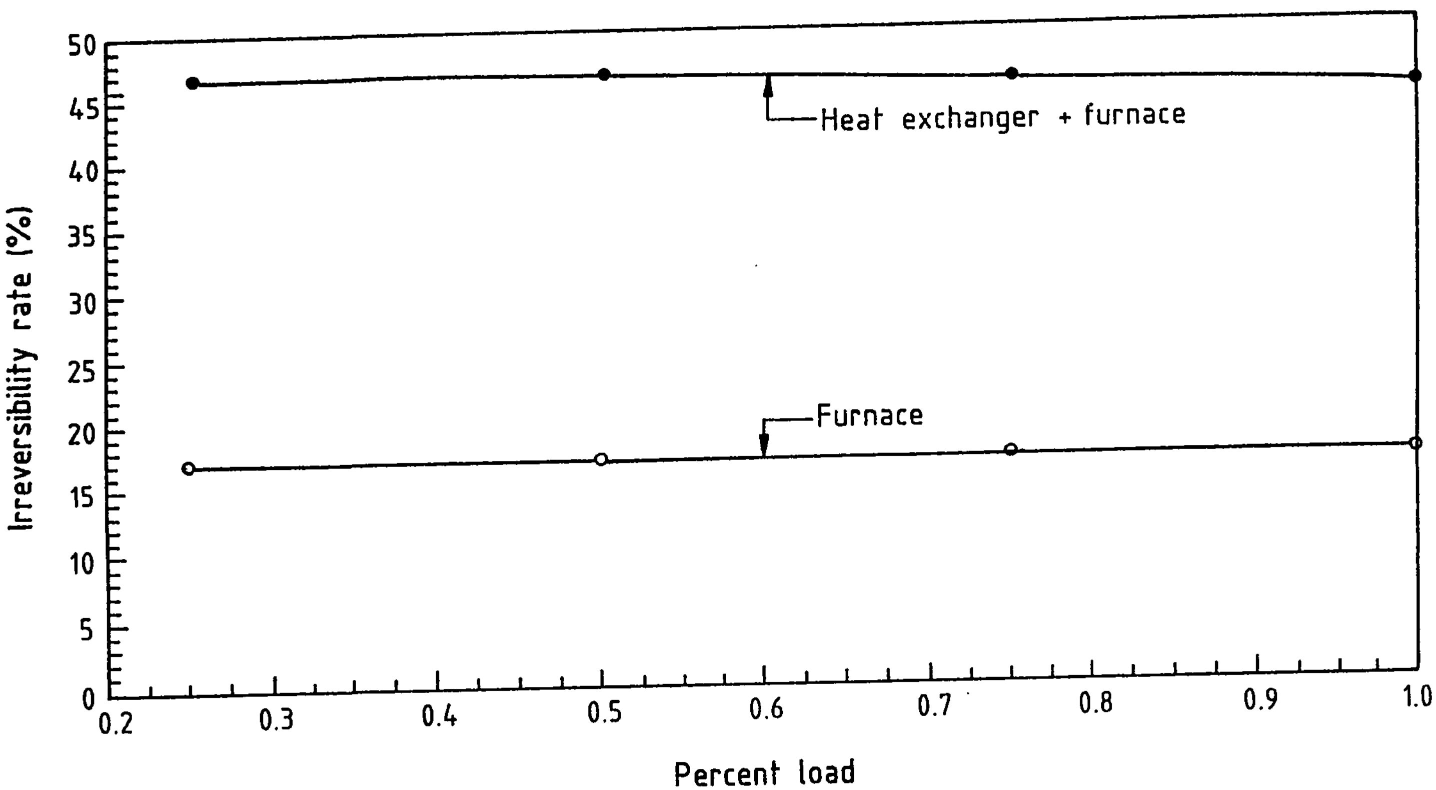


Fig. 4.5 Irreversibility rate versus percent load for heat exchanger and furnace.

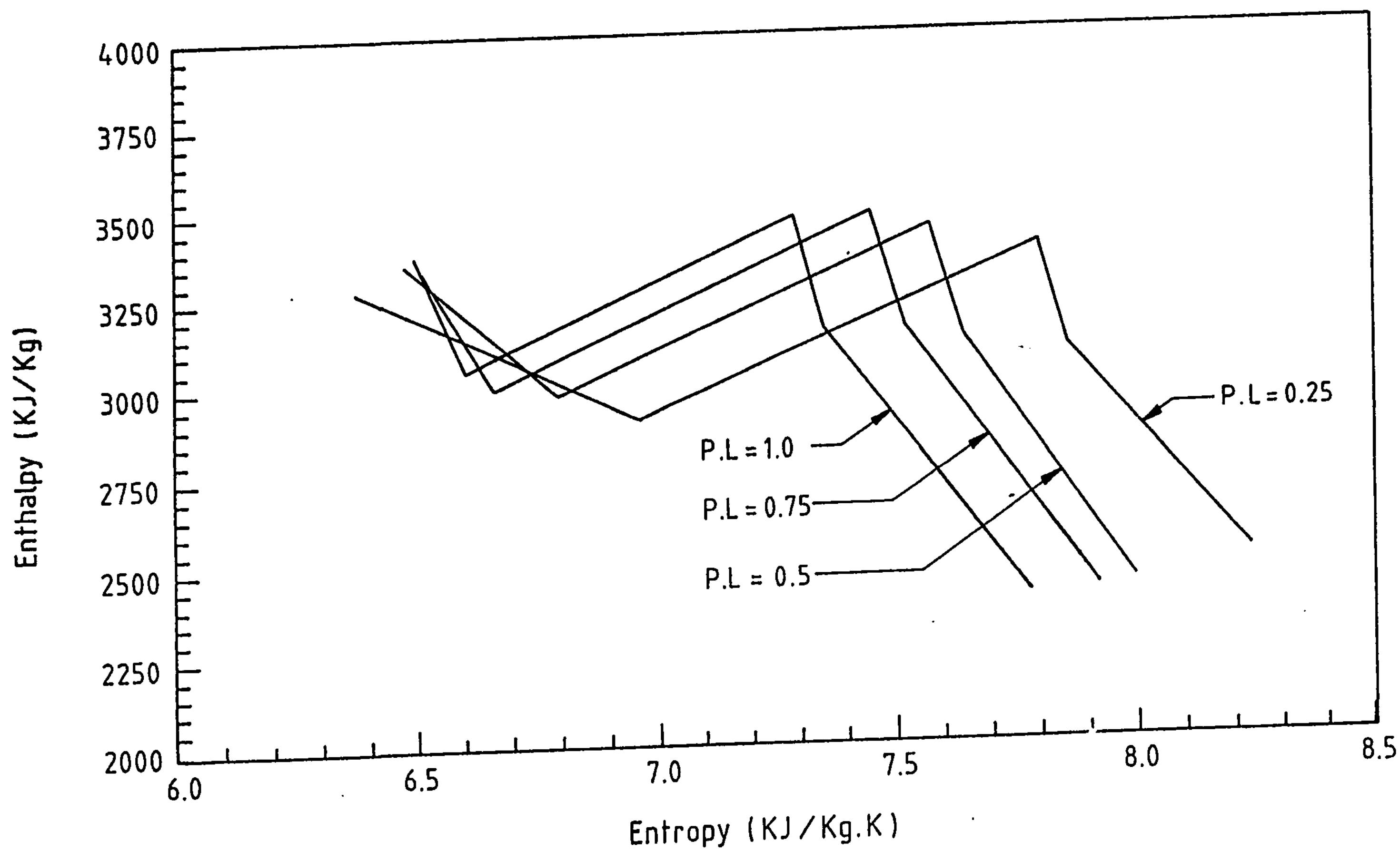


Fig. 4.6 Enthalpy versus entropy for different percent loads.

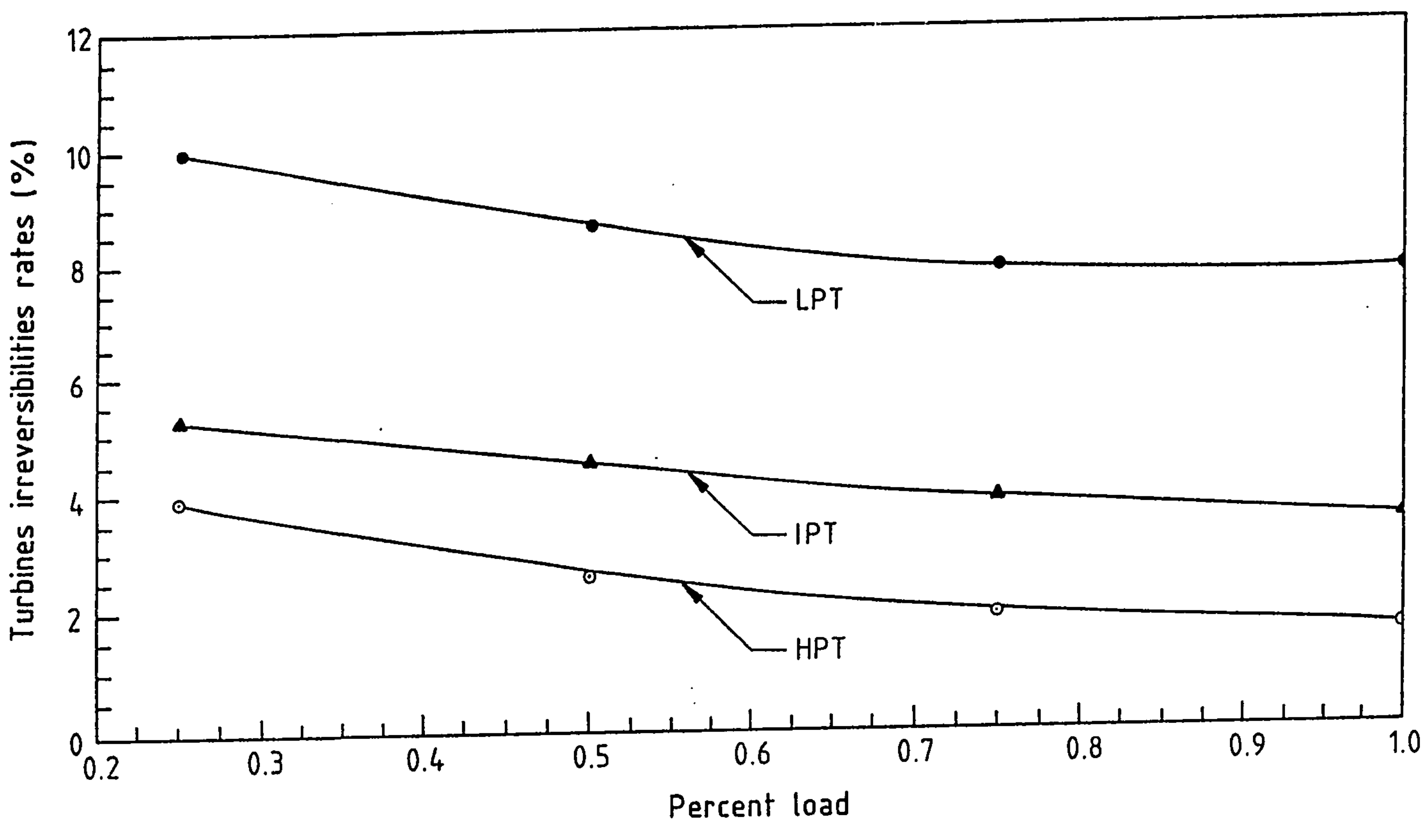


Fig. 4.7 Turbines irreversibilities versus percent load.

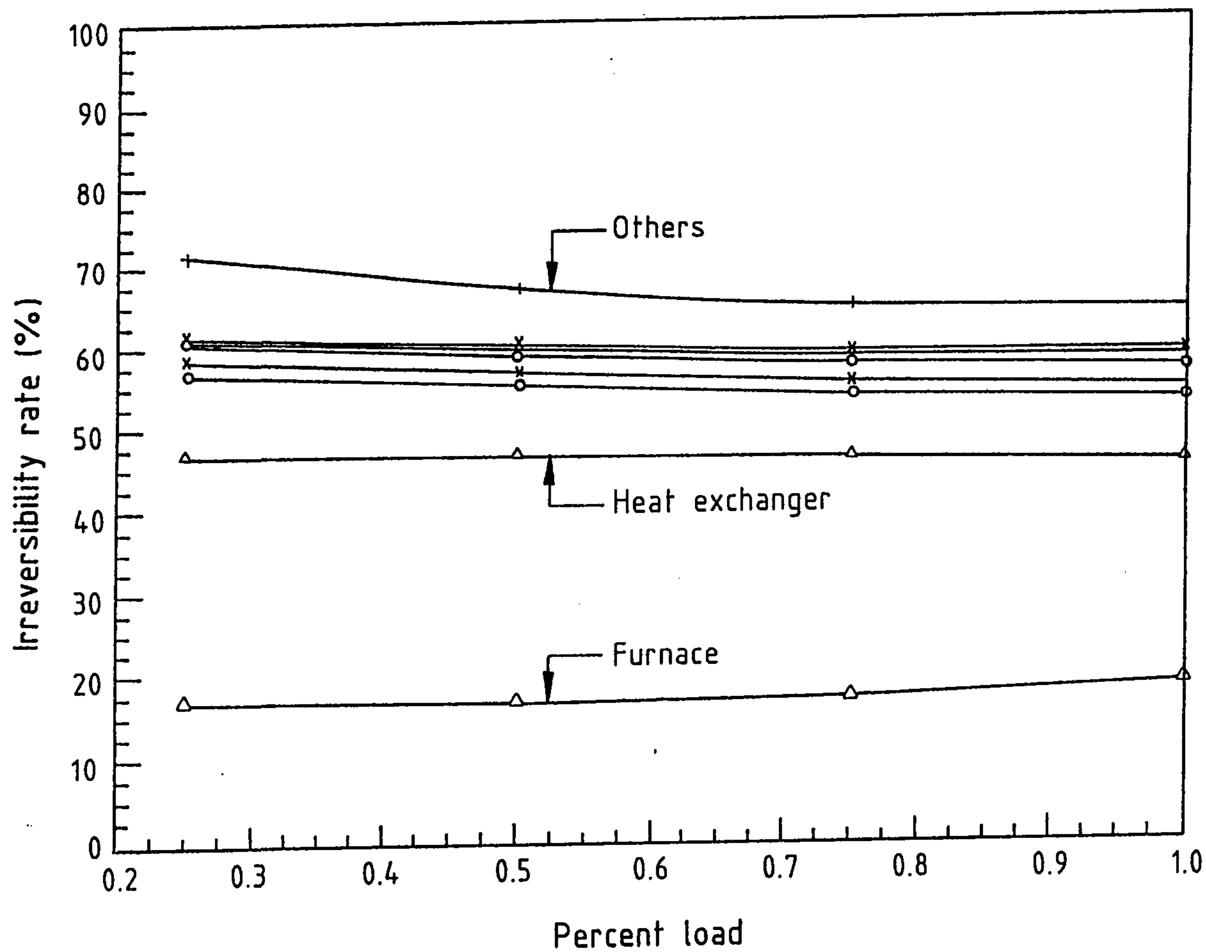


Fig. 4.8 Plant components irreversibilities versus percent load.

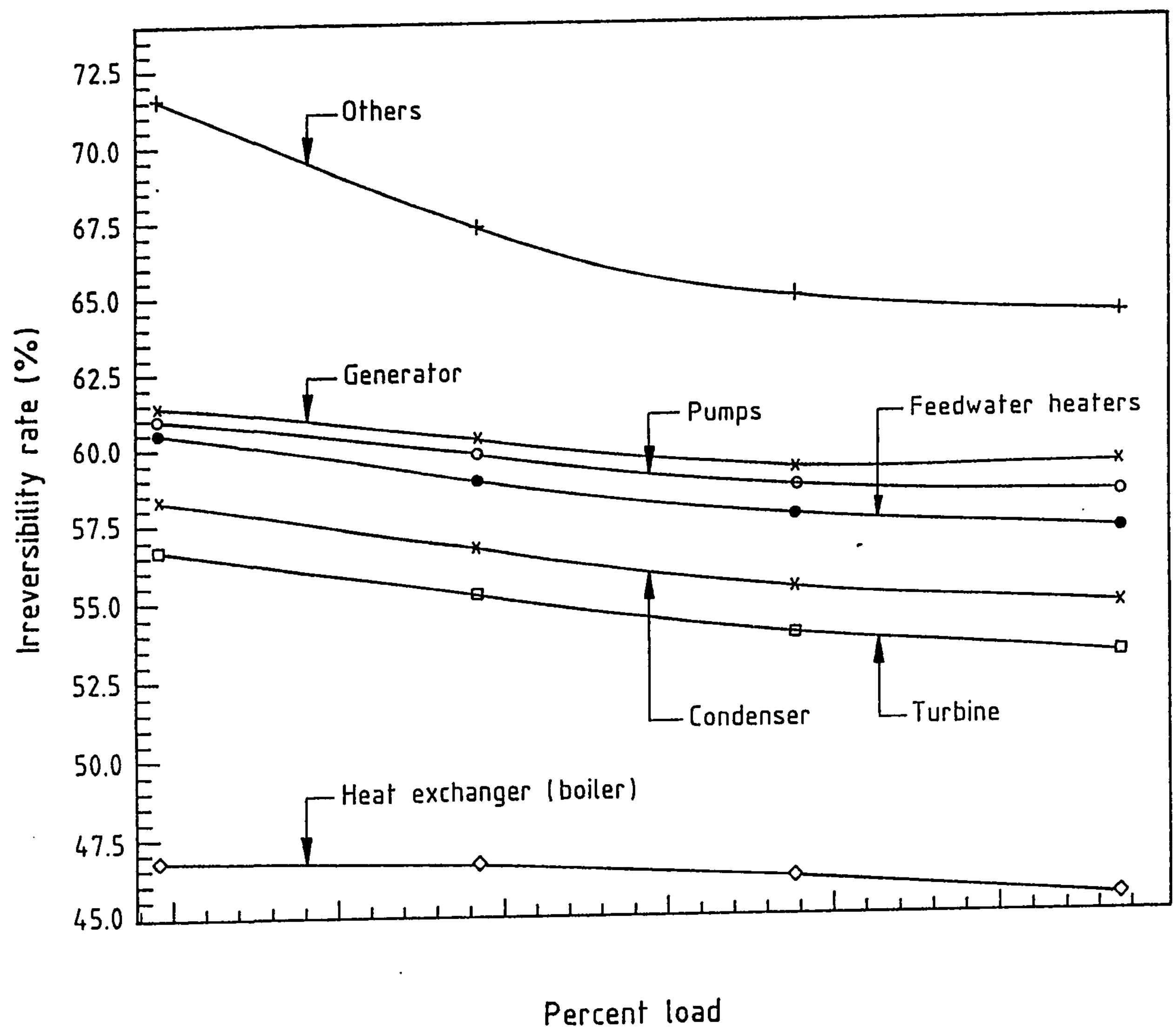


Fig. 4.9 Enlargement of Fig. 4.8 for irreversibility rate between 45% to 73% of the input exergy.

The results of the full load exergy analysis are shown in Fig. 4.10 and Table 4.2. The table shows that more than 70% of the total losses occur in the steam generator (Heat exchanger and furnace), 12% in the turbines, 2% in the condenser, 4% in the feedwater heaters, 2% in the pumps, and about 10% account for other losses such as friction, losses in fans, stack, etc.

Heat exchangers are major components in most energy systems, and they are the source of significant exergy losses. The high irreversibility in the heat exchanger is due to finite temperature difference between the combustion gases and the working fluid (water and steam). Heat exchangers are generally inefficient from an exergy standpoint because they have been designed in the past on the basis of low first cost that dictates a minimum sized unit. To achieve the small-sized heat exchanger, the temperature difference between the fluid streams is maximized. The larger is the temperature difference in a heat exchanger, the greater will be the exergy loss during heat transfer. A combustion process usually occurs simultaneously with heat transfer. Both the chemical reaction and the heat transfer are irreversible processes. The losses in the combustion chamber are due to the increase in the entropy of the combustion gases, heat lost with the products of combustion leaving the stack and due to incomplete chemical combustion. Exergy losses in turbine are due to frictional pressure drop in the steam pipes and across the governing throttle valve between the boiler and turbine. The enthalpy-entropy diagram (Fig. 4.6) is very useful in showing turbines performance characteristics and can be used to show the entropy differences in the turbines. The irreversibilities of high, intermediate and low pressure turbines are shown in Fig. 4.7. The 2% exergy losses in the condenser are due to heat transfer between the working fluid and cooling water which takes place at a finite temperature difference. The exergy losses in the pumps which are due to compression are usually small compared to the losses in the turbines. The exergy losses in the feedwater heaters are shown in Fig. 4.11.

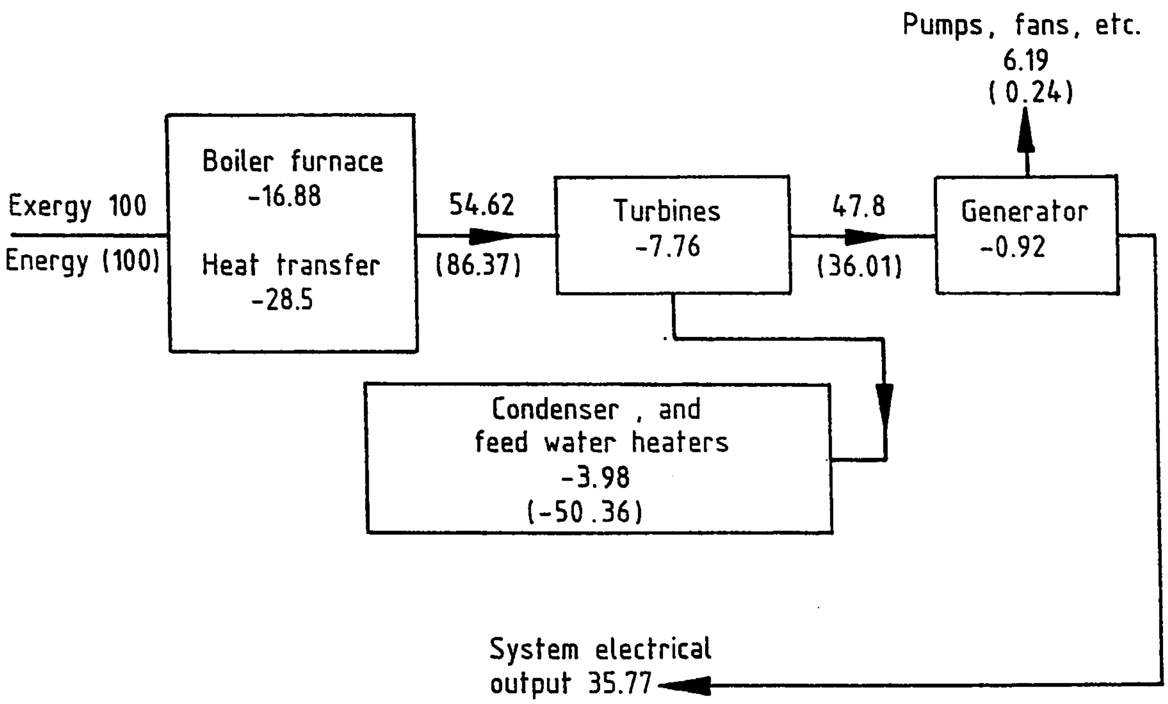


Fig. 4.10 Exergy and (energy) flow diagrams for design conditions.

Table 4.2. Exergy Analysis for Ghazlan Power Plant - Design Data

Percent load= 1.0
Input = 1080.8 MW
Output = 386.6 MW

	Exergy Loss MW	Losses as % of Input Exergy	Losses as % of Total Losses	Device Efficiency
SYSTEM	694.2	64.23	100.0	35.77
BOILER - furnace	182.43	16.88	26.24	86.37
BOILER - Heat Transfer	308.00	28.50	44.36	
Turbines	83.87	7.76	12.08	82.54
Generator	9.94	0.92	1.43	98.08
Condenser	15.89	1.47	2.29	
Feedwater Heaters	27.13	2.51	3.91	
Pumps	13.51	1.26	1.96	
Fans, Stack, etc.	53.43	4.93	7.70	

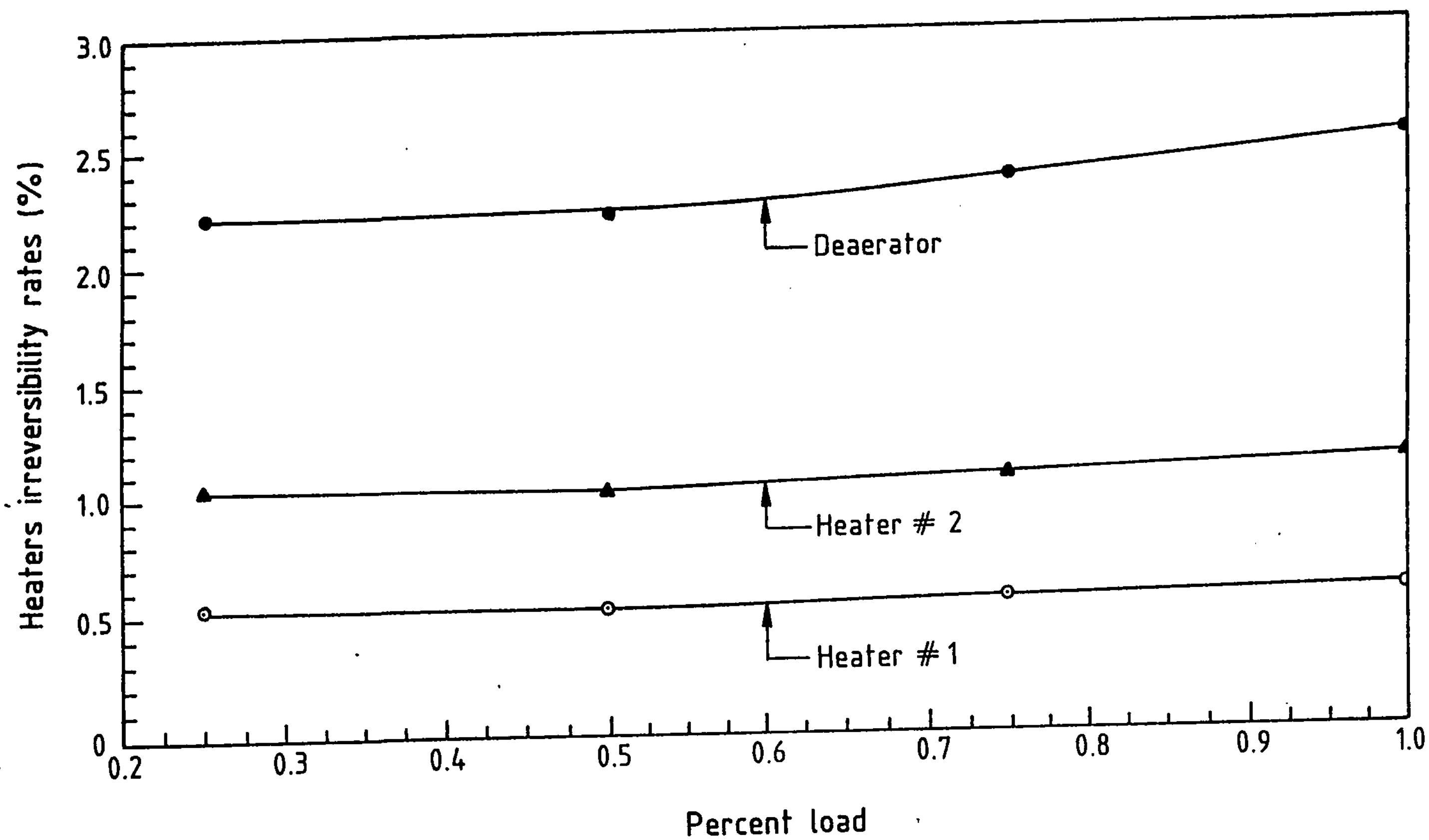


Fig. 4.11 Heaters irreversibility versus percent load.

The irreversibility of deaerator is higher than those of heater # 1 and heater # 2 because the extraction temperature and pressure for the deaerator is done at higher values compared to those for the closed feedwater heaters # 1 and # 2. This resulted in a temperature difference (between the extraction point and exit from the heater) of 155 ^{o}C for the deaerator, 104 ^{o}C for heater # 2 and 34 ^{o}C for heater # 1. Hence, this explains the higher irreversibility for the deaerator compared to others. Other losses which amount to about 10% of the total losses include friction in the plant piping system. The overall system efficiency, defined as the ratio of all of the useful exergy outputs to all the exergy inputs, is about 36%. The results reveal that for this particular plant, the heat transfer process is the most inefficient operation, accounting for nearly 44% of the total exergy destruction.

In Fig. 4.9, the energy and exergy values are displayed to compare the results of the first- and the second-law analysis of the plant. According to the first law analysis the major energy losses are due to the heat rejected in the condenser (50%) and with flue gases (14%); while according to the second law analysis the major losses are due to the boiler combustion chamber and the boiler heat exchanger in which 17% and 28% of the fuel exergy is destroyed, respectively. The first law results, if not interpreted correctly will give misleading results, therefore the power plant evaluation should be based on the second law results.

The second set of results pertains to the comparison between the design and actual performance of the plant. Figure 4.12 contains the results of the overall efficiency and total irreversibilities of the plant as a function of load. The actual efficiency is (on average) about 4.0% less than the corresponding design ones while the actual total irreversibility is (on average) about 4.0% higher than the corresponding design ones depending on the percent load. As can be noted from the figure, the available plant actual data covers a load ranging from 250 MW to 340 MW. The comparison results reveal that there is a room for

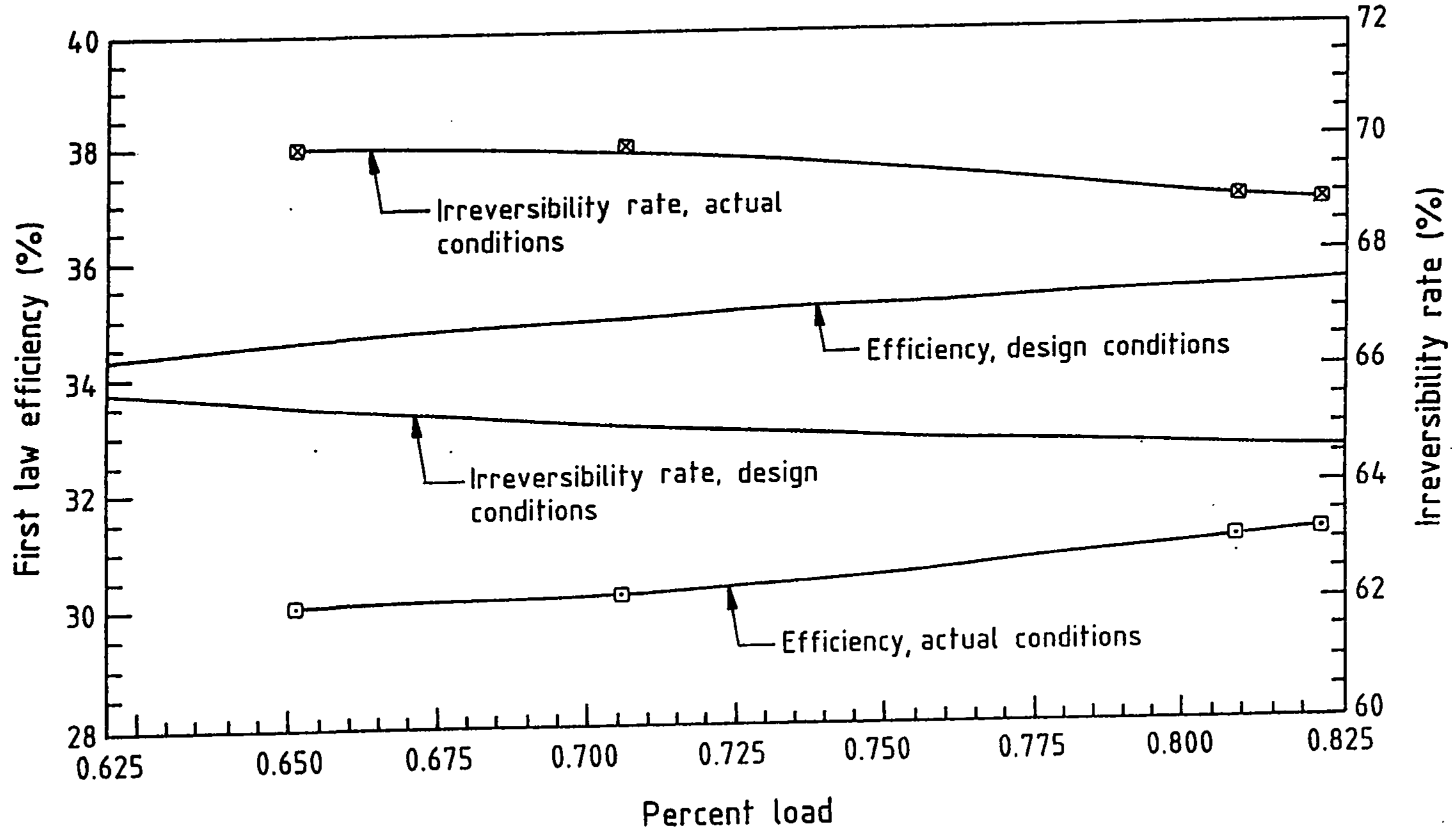


Fig. 4.12 Efficiency and irreversibility rate versus percent load for design and actual conditions.

improvement in the actual plant performance. The results of the energy and exergy analysis at full load are shown in Fig. 4.13 and Table 4.3. It can be seen that the major inefficiencies of the plant are located in the boiler heat exchanger. A parametric study will be conducted to quantify the improvements that can be achieved, in the actual performance of the plant, by considering different measures. The results of the parametric study is the subject of the following discussion.

The third set of results pertains to the comparison between the full load design performance of the plant under study and an existing 600 MW Qurayyah fuel-fired electrical plant with 8 feedwater heaters. The results of the exergy analysis of the 600 MW plant is given in Table 4.4. The results shown in Tables 4.2 and 4.4 indicate a difference of 7% in the system efficiencies with the plant under study having the lower efficiency. Part of this difference could be attributed to the number of feedwater heaters used.

4.2 ECONOMICS OF ELECTRIC POWER PRODUCTION

One of the principal components of electric power cost to the consumer is the generation cost. The generation cost generally represents about 60% of the total cost of electric power to the consumer and is the main concern of power engineers responsible for power plant economics. One of the three major cost elements of generation is the fuel cost. For fossil-fueled power plants, this includes the cost of gas, petroleum or coal and interest charges on fuel reserve required at plant sites. In this section the savings in fuel cost for one year will be evaluated for 1% improvement in the overall plant efficiency. The improvements can be achieved by several alternative arrangements such as: increasing number of feedwater heaters and increasing the throttle steam temperature.

To evaluate the fuel cost, the overall plant efficiency, the capacity factor, the fuel higher heating value and the fuel price must be estimated. The fuel cost is given as

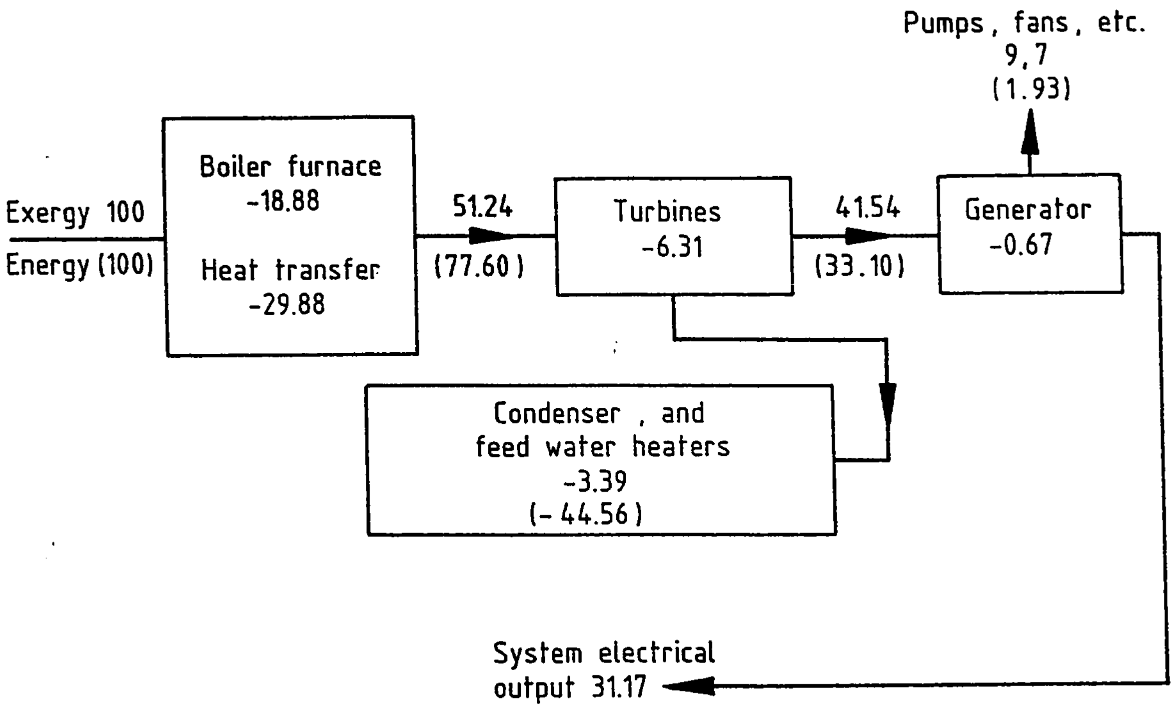


Fig. 4.13 Exergy and (energy) flow diagrams for actual operating conditions.

Table 4.3. Exergy Analysis for Ghazlan Power Plant - Actual Data

Input = 1080.8 MW
Output = 386.6 MW

	Exergy Loss MW	Losses as % of Input Exergy	Losses as % of Total Losses	Device Efficiency
SYSTEM	702.43	68.83	100.0	31.17
BOILER - furnace	192.67	18.88	24.73	77.6
BOILER - Heat Transfer	304.93	29.88	43.41	
Turbines	64.40	6.31	9.17	83.16
Generator	6.84	0.67	0.97	97.9
Condenser	12.65	1.24	1.8	
Feedwater Heaters	21.94	2.15	3.12	
Pumps	8.58	0.84	1.22	
Fans, Stack, etc.	90.42	8.86	12.87	

Table 4.4. Exergy Analysis for Qurayyah Power Plant

Percent load = 1.0
Input = 1436.5 MW
Output = 613.95 MW

	Exergy Loss MW	Losses as % of Input Exergy	Losses as % of Total Losses	Device Efficiency
SYSTEM	822.55	57.26	100.0	42.74
BOILER -furnace	242.48	16.88	29.47	88.10
BOILER - Heat Transfer	361.89	25.19	43.99	
Turbines	61.30	4.26	7.45	91.00
Generator	6.24	0.43	0.75	99.00
Condenser	32.07	2.23	3.89	-
Feedwater Heaers	29.99	2.08	3.64	-
Pumps	5.72	0.40	0.70	-
Fans, Stack, etc.	82.85	5.76	10.07	-

$$\text{Fuel Cost} = \frac{\dot{W} * C_p}{\eta_o \, x \, HHV} \text{ x } 8760 \text{ x } 3600 \left(\frac{kg}{yr}\right) \text{x C} \tag{56}$$

where $\dot{W}$ = gross power output in KW

C_p = plant capacity factor

η_o = overall plant efficiency

HHV = Fuel higher heating value kJ/kg

C = fuel price (SR/kg)

Using in equation (56) the following values:

$\dot{W} = 350,000$ KW

$C_p = 0.9$

η_o improved from 0.31 to 0.32

HHV = 50607 KJ/kg [31]

C = 0.387 SR/kg [30]

The annual savings in the fuel cost for 1% improvement in efficiency will be around 6 millions Riyals.

4.3 PARAMETRIC STUDY

The results of the comparative study indicated that, there is a room for improvement in the plant performance. The incentive for improving performance is enormous: a 1% change in performance represents millions of Saudi Riyals in operating costs as has been reported in the previous section. Hence, this section deals with the potential for improving the plant efficiency. Several alternative arrangements to improve the efficiency will be considered. The results of analyses using both the first- and second law of thermodynamics for the turbine cycle of the plant operating under various conditions will be presented. The physical quantities whose effects were examined are: throttle steam pressure and temperature, reheat steam pressure and temperature, effect of number of reheats and effect of number of feedwater heaters. Although the throttle steam conditions have been progressively tending toward higher temperatures and pressures, they have never been formally standardized. The literature [18] indicates that certain steam conditions have been selected more often than others. Based on practical considerations, the throttle steam temperatures selected in this study range from 670°K to 1000°K, and pressures vary from 9 to 18 MPa to suit the existing steam generation. The parametric studies in the following sections are made for fixed power which is the power at full load and the other conditions are fixed to the values shown in Table 4.1.

4.3.1 Effect of Throttle Steam Pressure

It is commonly reported that an increase in the throttle pressure improves the performance of the plant. Hence a parametric study was conducted to investigate the effect of the steam throttle pressure on the performance. Since the throttle steam pressure affects steam pressures at each extraction stage, and these extraction pressures also effect the performance of the cycle, the ratio of the extraction pressures to throttle pressure for the base case (design) was maintained for all cases. The results are shown in Fig. 4.14 for two throttle steam temperatures of 500 and 580°C. The results in the figure show that the effect of increasing throttle steam pressure is steady increase of the cycle first- law efficiencies. However, the rate of increase is not constant. The second law analysis shows a rather different trend. Figure 4.14 indicates that for the 500°C throttle steam temperature curve, the second law efficiency increases at first, when throttle steam pressure increases from 9

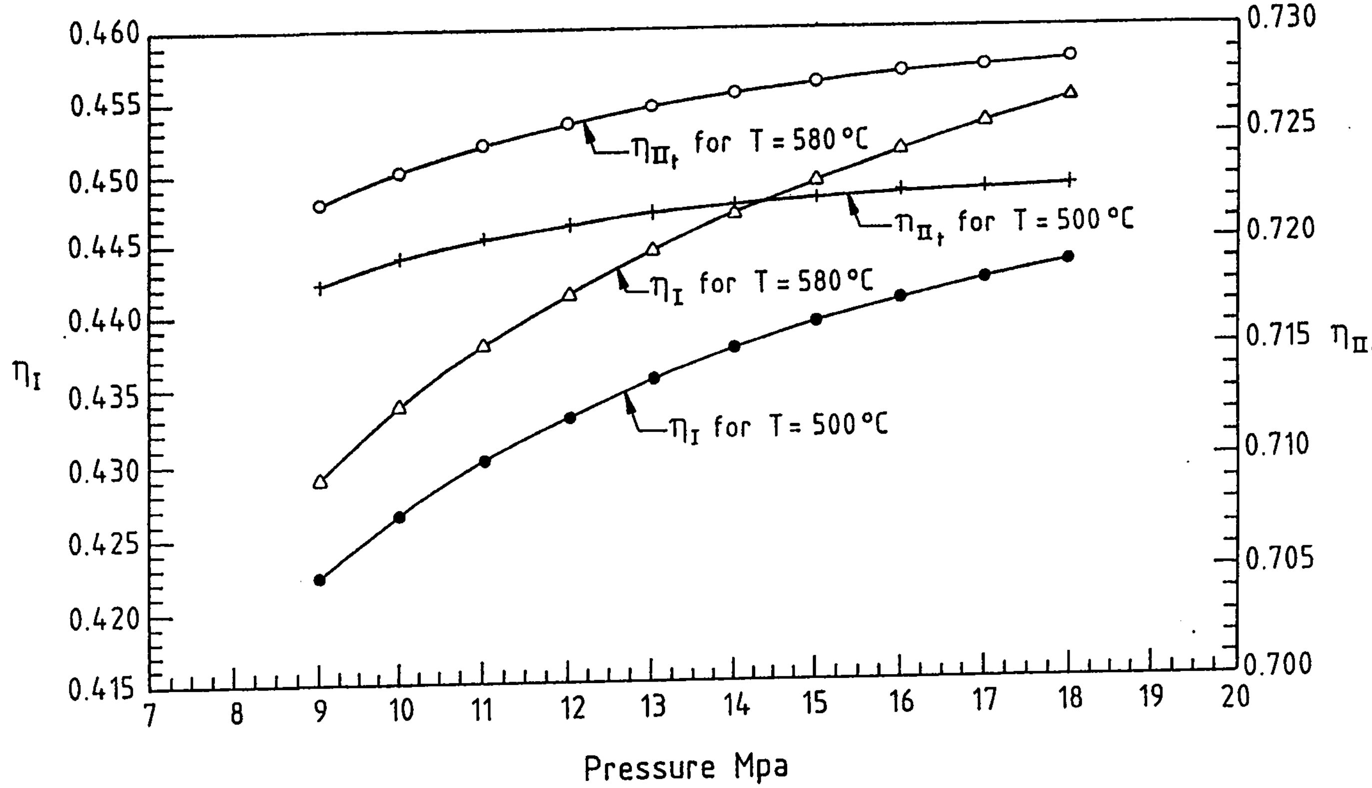


Fig. 4.14 · Effect of throttle steam pressure on the performance

MPa to 15 MPa. Further increase of throttle pressure from 15 MPa to 16.5 MPa improves the second law efficiency slightly, and for throttle pressures above 16.5 MPa (and up to 18 MPa in this calculation) the second law efficiency remains constant. The 580°C throttle steam temperature curve exhibits a similar trend with a slight difference. Hence, increasing the throttle steam pressure from 12 MPa (present value) to 18 MPa will improve the turbine cycle first law efficiency by 1.07% and the second-law efficiency by 0.17% for 580°C throttle steam temperatures. Figures 4.15 and 4.16 for 500°C and 580°C throttle temperatures, show that the second law efficiency increases slightly with the increase of throttle pressure, however, the fuel flow rate decreases. Therefore, for certain throttle temperature, the use of high throttle steam pressure should be evaluated on the basis of the second law analysis to determine if it is possible to even obtain a gain in performance by increasing it.

4.3.2 Effect of Throttle Steam Temperature

The Rankine cycle efficiency analysis indicates that the higher the throttle steam temperature, the higher is the cycle efficiency. Hence, a parametric study was conducted to investigate the effect of throttle steam temperature on the plant performance. The significance of the gains in performance has been realized and because of the incentive that will be gained, efforts are continuing to overcome the material limitations. Figure 4.17 shows the effect of throttle steam temperature on the cycle performance. Both the first- and second-law efficiencies increase with increasing throttle temperature. However, the rate of increase is different. The gain in the first-law efficiency is about 1.05%, if the steam temperature increases from 785°K to 885°K or per 100°K. For the second law efficiency the gain is about 0.6% for an increase from 785°K to 885°K in steam temperature or per 100°K.

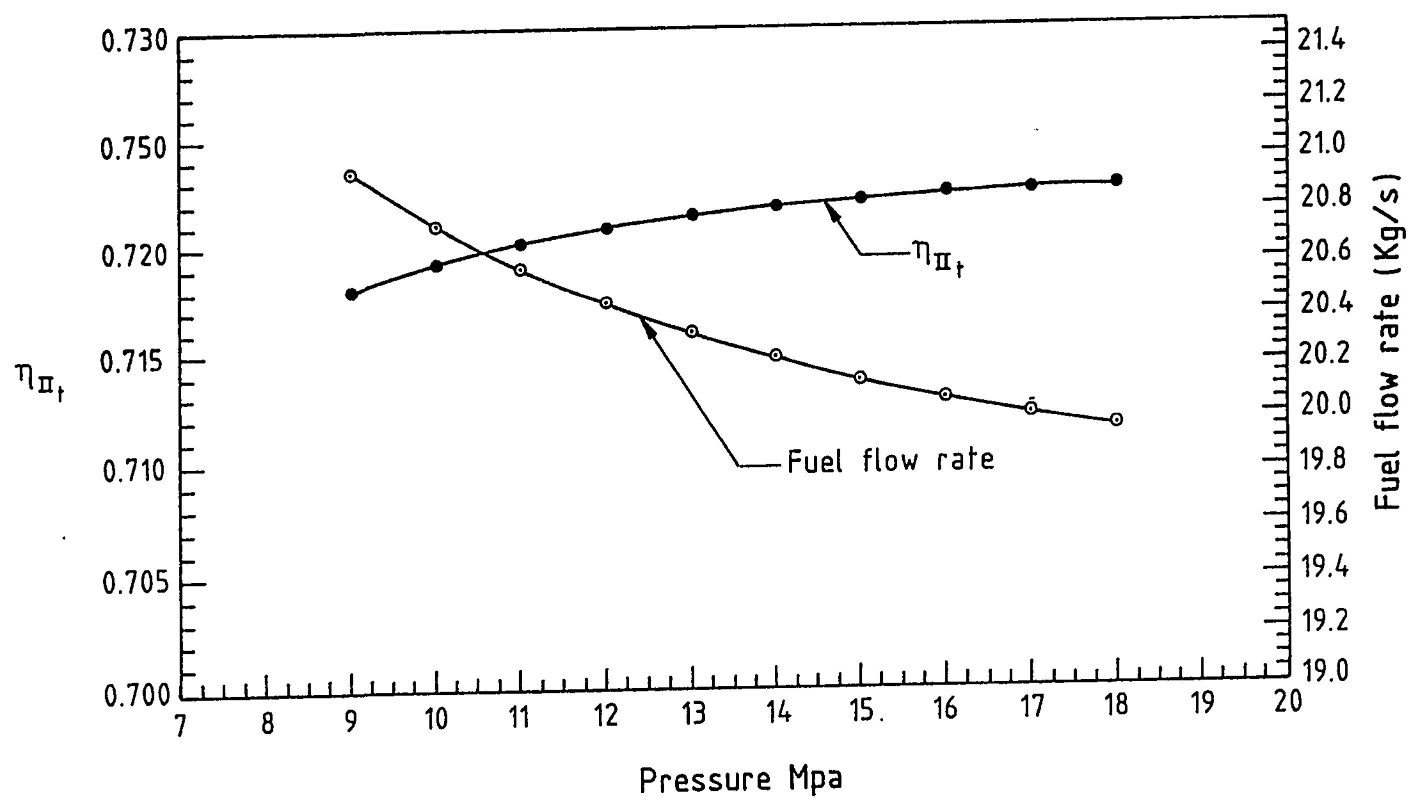


Fig. 4.15 Effect of throttle steam pressure on the performance and fuel flow rate for throttle temperature = 500 °C

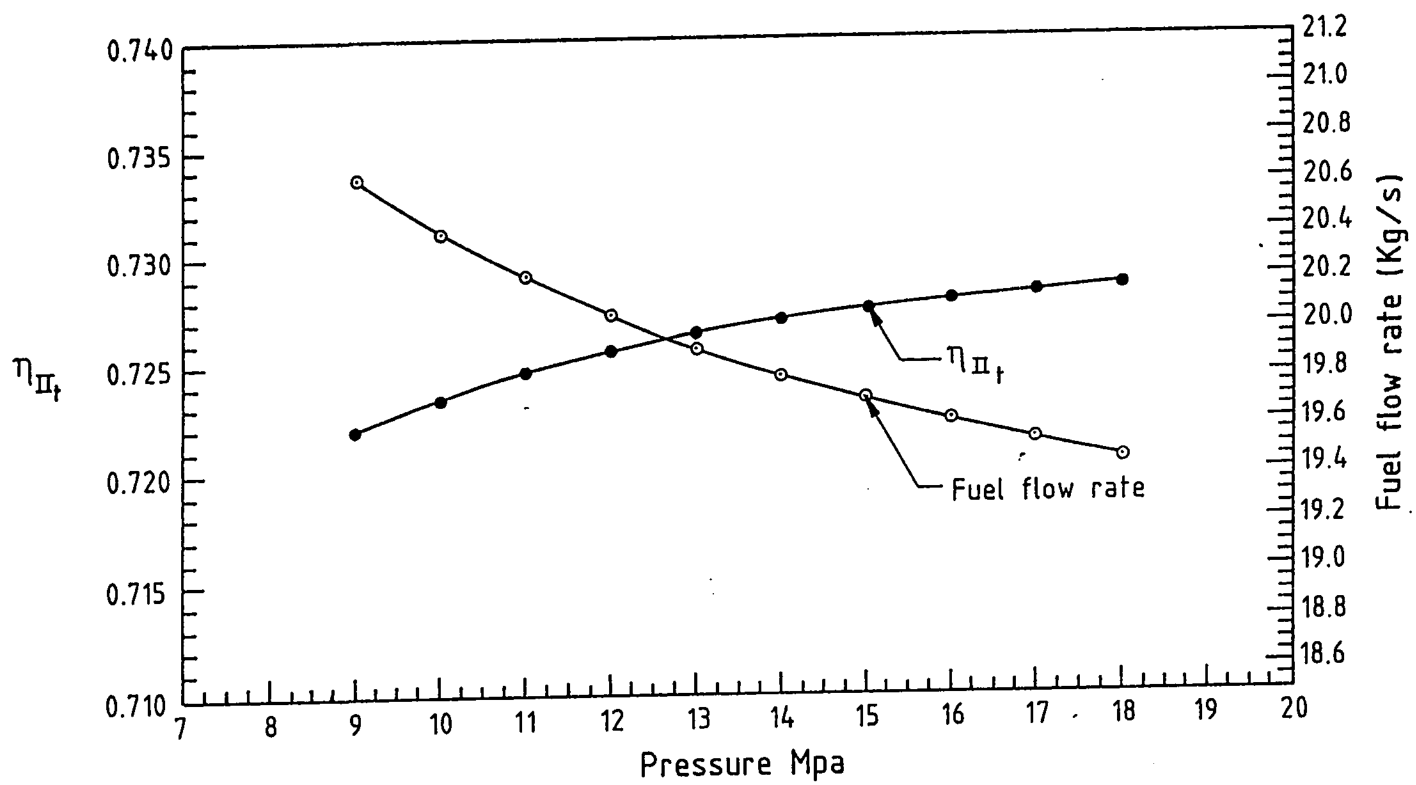


Fig. 4.16 Effect of throttle steam pressure on the performance and fuel flow rate for throttle temperature = 580 °C

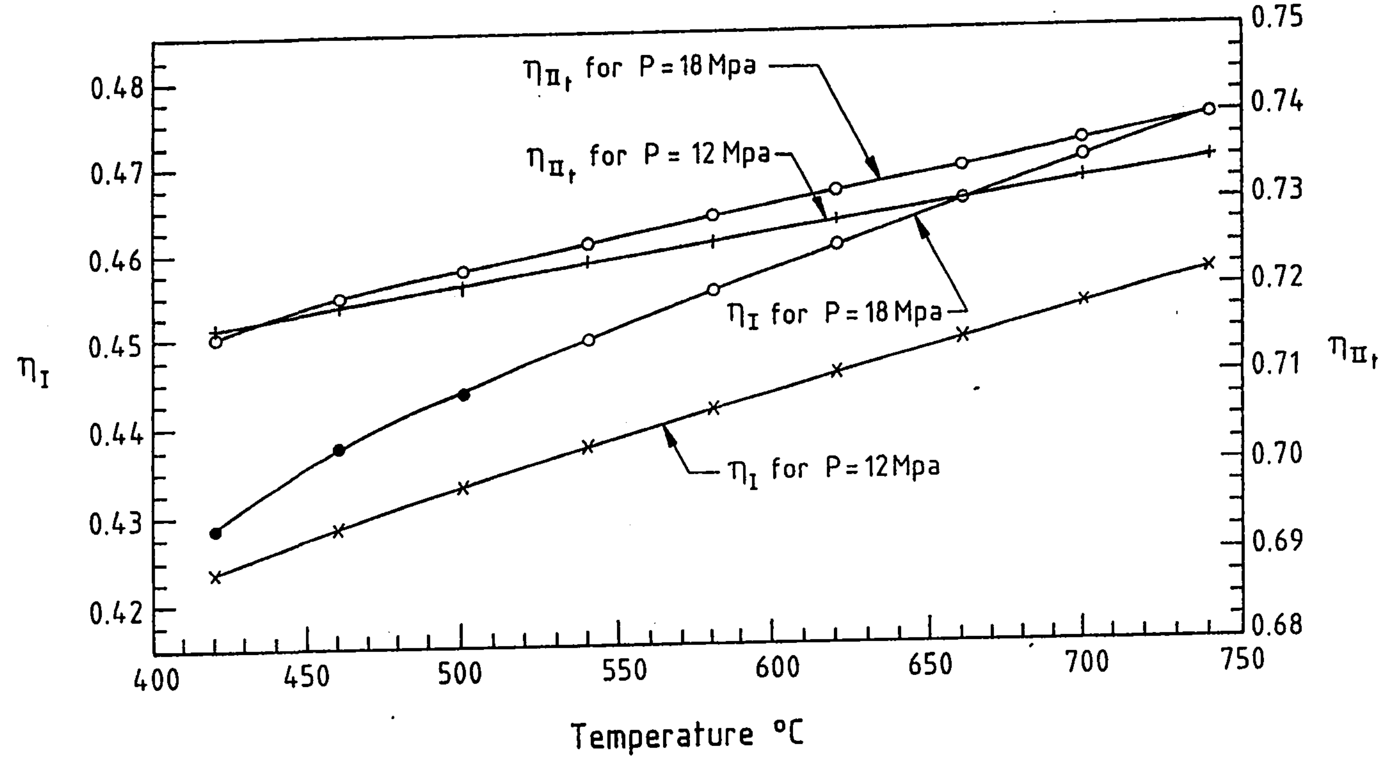


Fig. 4.17 Effect of throttle steam temperature on the performance.

4.3.3 Effect of Reheat Steam Pressure For a Single Reheat

A parametric analysis was conducted to investigate the effect of single reheat steam pressure on the performance of the cycle. The results as shown in Fig. 4.18 for ratios of reheat pressures to throttle pressure of zero to 1. The results indicate that for every set of steam conditions an optimum reheat pressure exists which yields maximum efficiency. The study shows that the optimum reheat pressure is 3.83 MPa, however, the single reheat pressure used is around 2.85 MPa.

4.3.4 Effect of Reheat Steam Temperature For a Single Reheat

Reheating is a common design feature of modern large power plants. The results in Fig. 4.19 show a steady increase in the first-law efficiency with increasing reheat temperature. The rate of improvement is about 0.46% per 100^{o}K in the degree of reheat temperature range of 66^{o}C to 166^{o}C and 0.42% per 100K in the range of 166-316^{o}C. The second-law analysis shows an increase in second-law efficiency which is substantially different - the efficiency increases sharply in the degree of reheat range of 66^{o}C to 276^{o}C but becomes almost constant at about 71.33% for degree of reheat greater than 276^{o}C. This fact has important practical implications. The steady increase of the first-law efficiency has routinely provided an economic incentive for the use of higher temperature. The second-law analysis gives the optimal reheat temperature which is about the throttle steam temperature. For this reason, single or double reheat about the same temperature as the initial throttle steam temperature is commonly used in modern larger power plants.

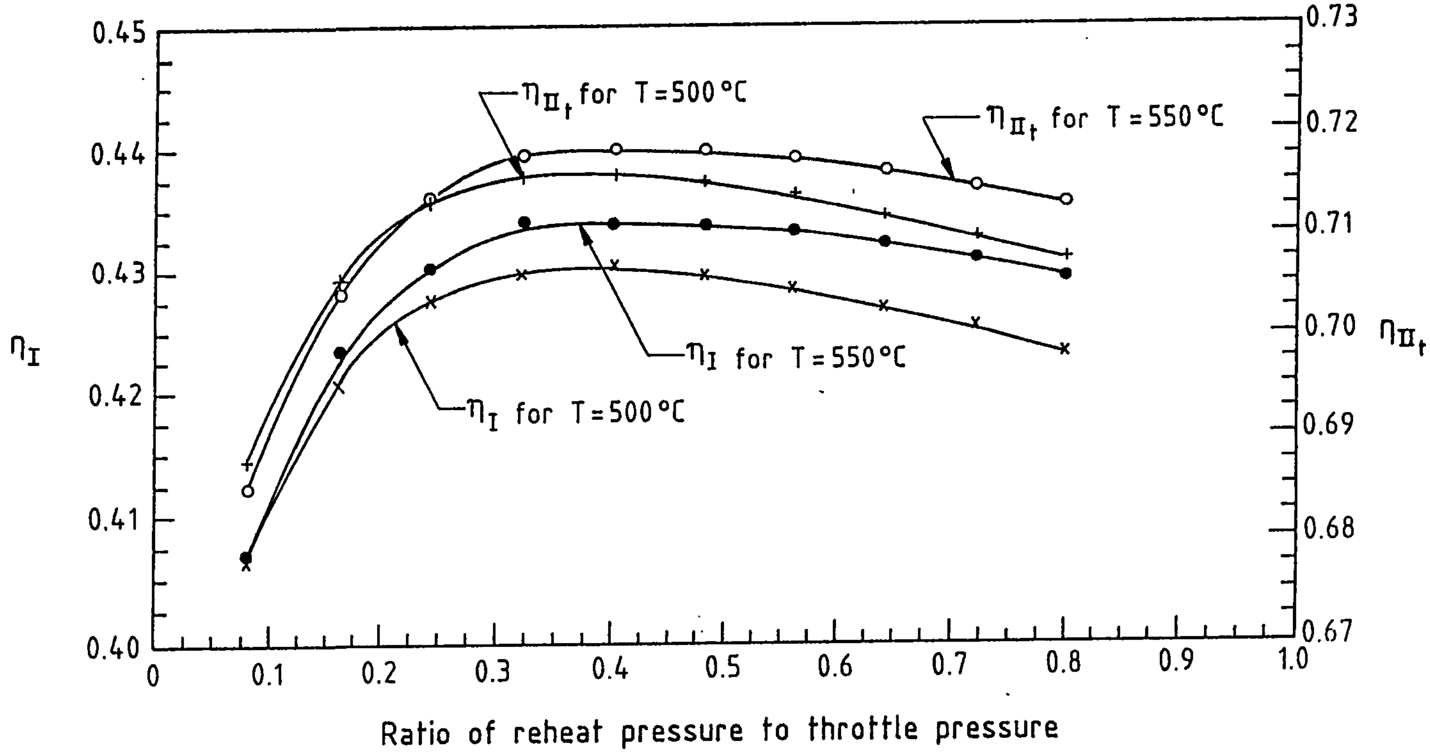


Fig. 4.18 Effect of reheat steam pressure on the performance.

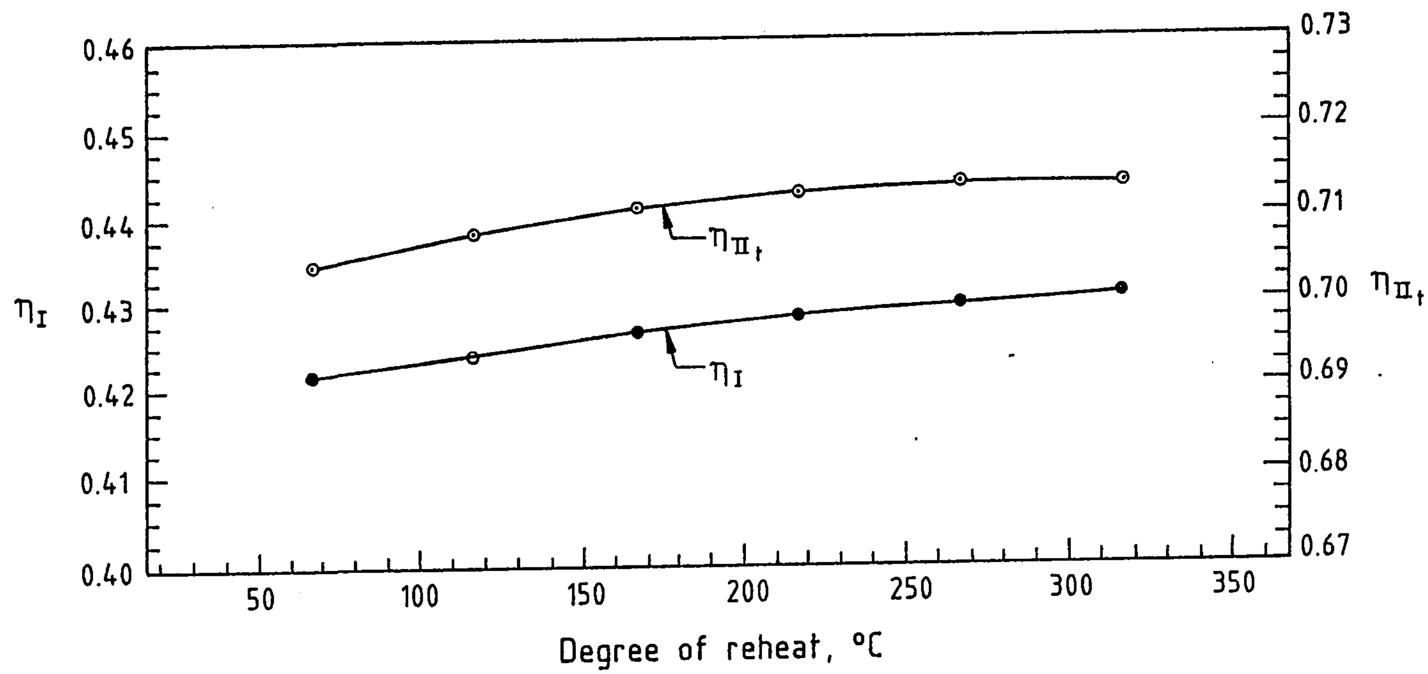


Fig. 4.19 Effect of reheat steam temperature on the performance.

4.3.5 Effect of a Double Reheat

To study the effect of a double reheat on the cycle performance, a detailed first- and second-law analyses were carried out to determine the optimum reheat pressures for each reheat. The result of this analysis is shown in Figs. 4.20 to 4.23.

In Fig. 4.20, the abscissa represents the ratio of reheat # 2 pressure to the throttle pressure, the ordinate represents the first-law efficiency and the different curves represent different ratios of reheat # 1 pressures over the throttle pressure. The solid line represent the limiting case. Based on the first-law analysis, the optimal pressures were obtained using Fig. 4.20 with the reheat # 1 pressure being 24% of the throttle pressure and reheat # 2 pressure being 4.5% of the throttle pressure. Figure 4.21 gives the second-law efficiency at different reheat # 1 and reheat # 2 pressures. The cycle total irreversibility rate at different reheat # 1 and reheat # 2 pressures is shown in Fig. 4.22. The second-law analysis results yield the same optimum reheat pressures.

The results in Fig. 4.23 show that increasing the number of reheat does increase the cycle performance. The first-law efficiency increased by 1% and the fuel flow rate decreased by 0.45 kg/s with a double reheat compared to a single reheat option.

4.3.6 Effect of Number of Feedwater Heaters

One way of improving the performance of this existing plant is to increase the number of feedwater heaters. The number of feedwater heaters used in modern power plants could be as high as 8, while only 3 are used in this plant. The optimum gain due a single heater is about one-half of that due to an infinite number of heaters. This account for the fact that the number of heaters actually installed does not need to be more than that given in Table 4.5 [27]. As shown in Fig.4.24, the efficiency increases as the number of

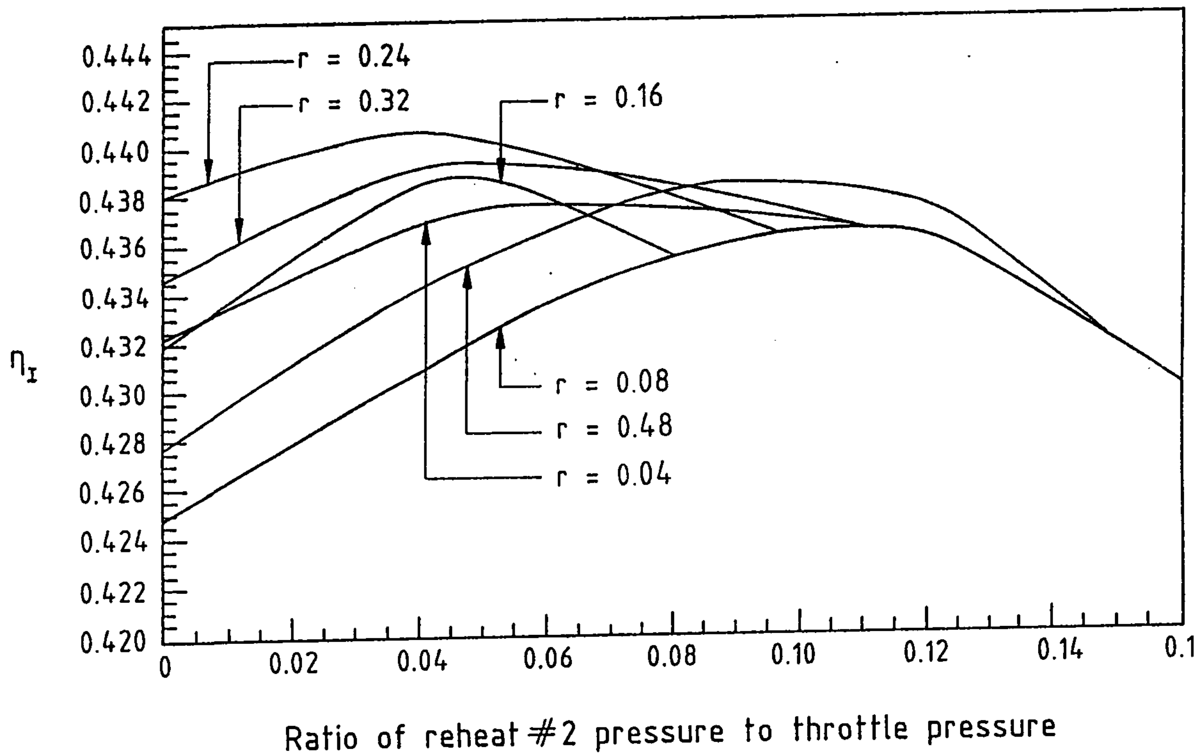


Fig. 4.20 Effect of reheat #1 and #2 pressures on the first law efficiency.

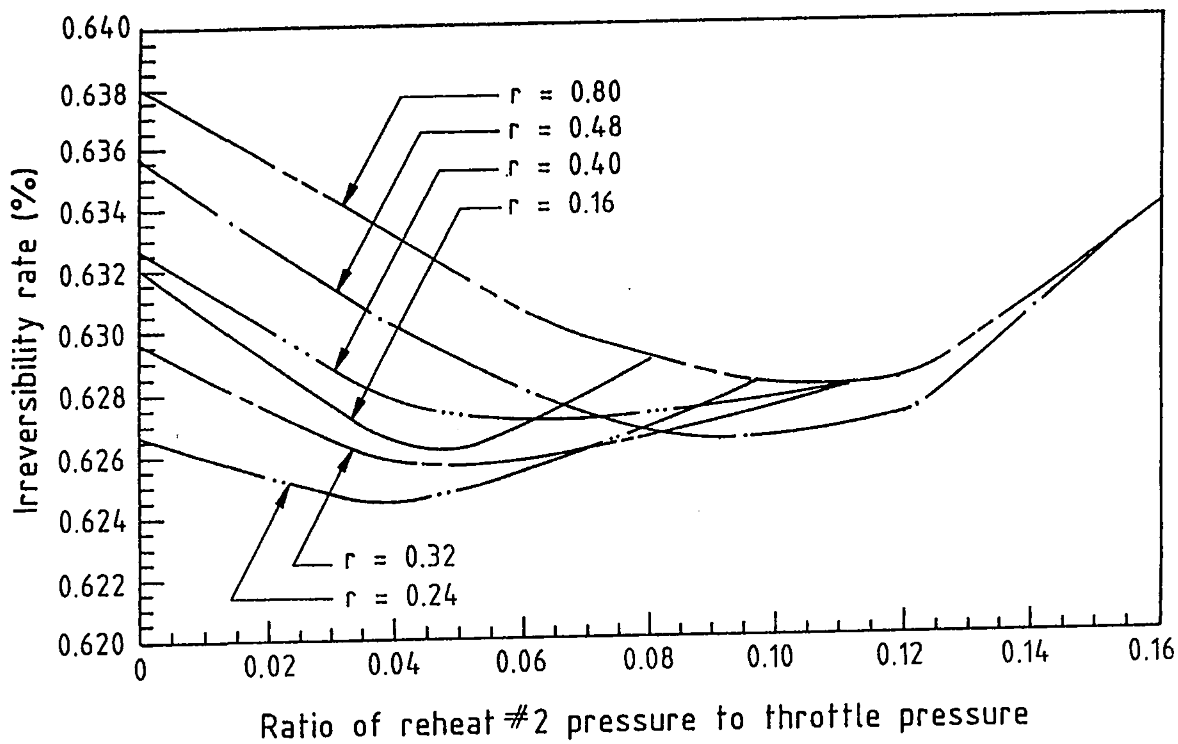


Fig. 4.21 Effect of reheat #1 and #2 pressures on the total irreversibility.

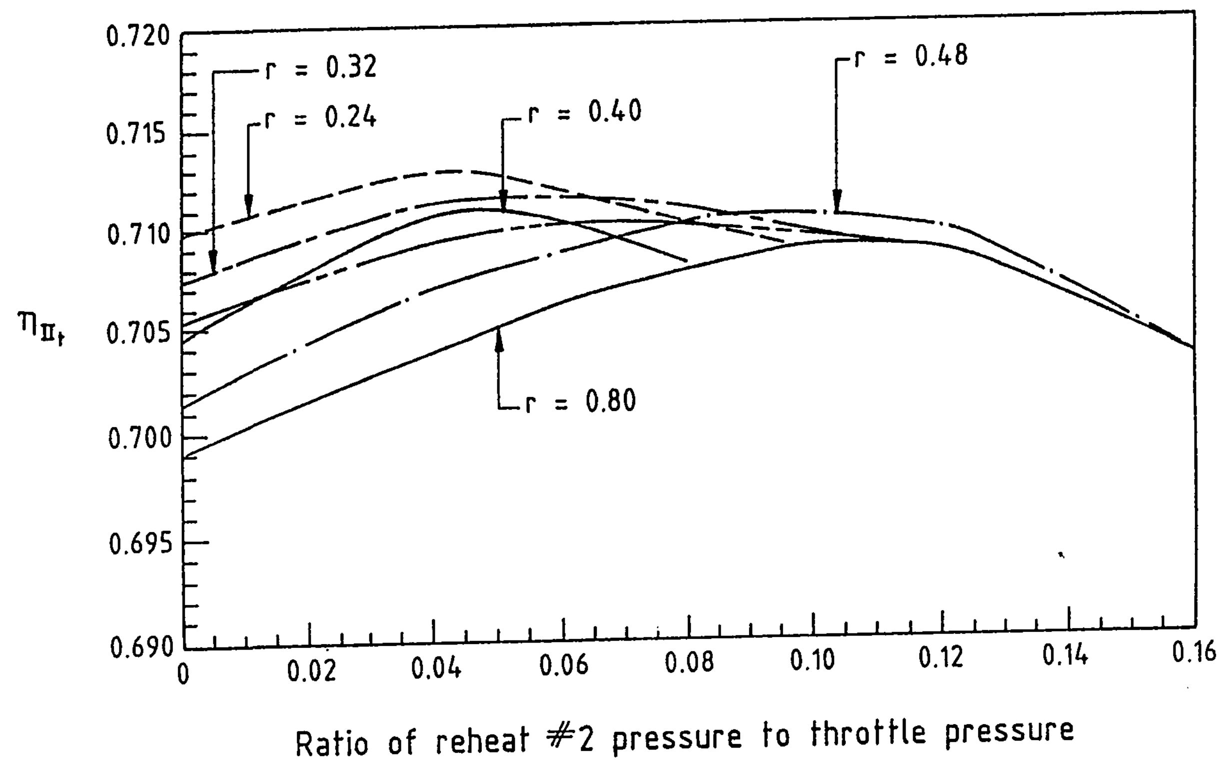


Fig. 4.22 Effect of reheat #1 and #2 pressures on the second law efficiency.

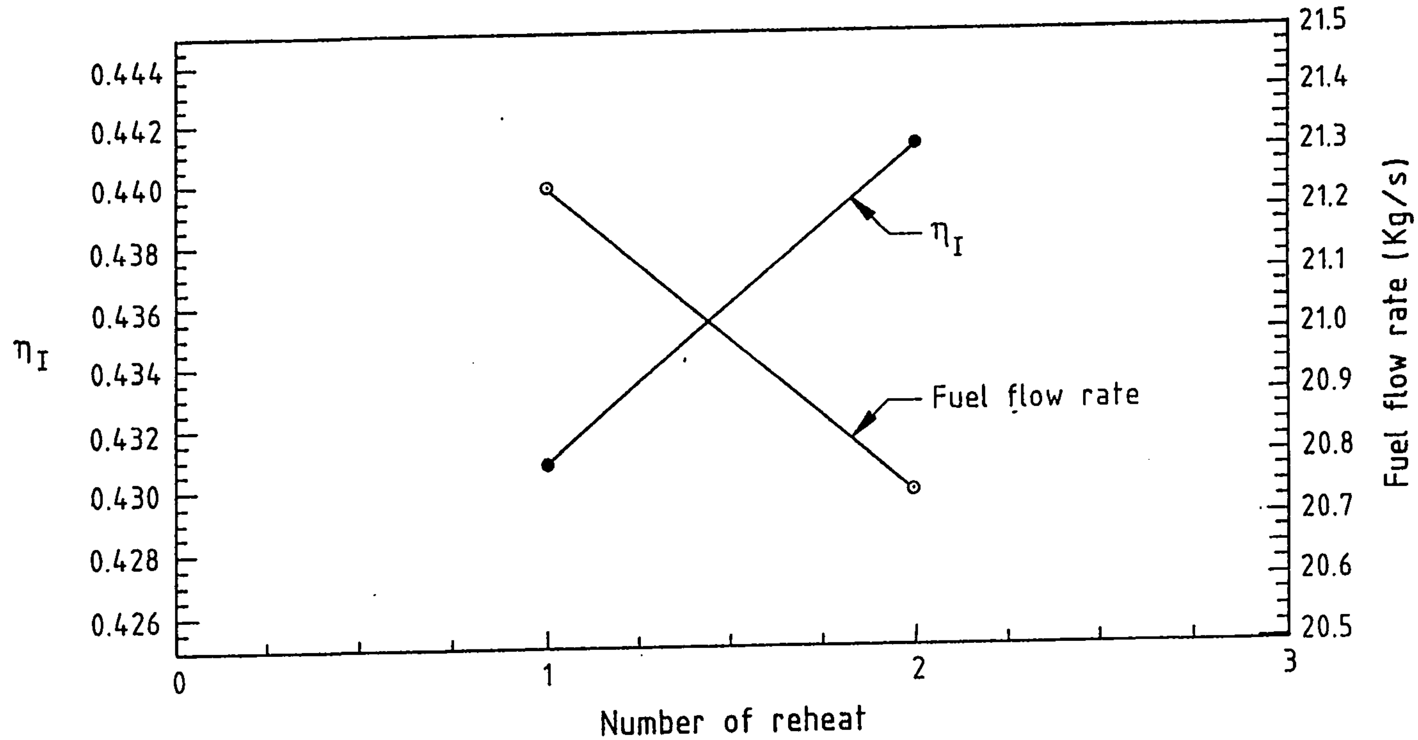


Fig. 4.23 Effect of number of reheat on the performance and fuel flow rate at optimum reheat pressures.

Table 4.5. Development in Power Plant Peformance [27].

Approximate Year of installation		United Kingdom								Supercritical pressure machines			
										U.K	USA		
		1907	1919	1938	1950	1958	1959	1966	1973	1965	1959	1972	1975
Output	MW	5	20	30	60	120	200	500	660	375	450	800	1300
Gauge pressure	lbf/in^2 (MN/m^2)	190 (1.3)	200 (1.4)	600 (4.1)	900 (6.2)	1500 (10.3)	2350 (16.2)	2300 (15.9)	2300 (15.9)	3500 (24.1)	3500 (24.1)	3500 (24.1)	3500 (24.1)
Initial temp.	^{o}F (^{o}C)	500 (260)	600 (316)	850 (454)	900 (482)	1000 (538)	1050 (566)	1050 (566)	(1049) 565	1100 (593)	1050 (566)	1000 (538)	1000 (538)
Ist reheat temp.	^{o}F (^{o}C)	-	-	-	-	1000 (538)	1000 (538)	1050 (566)	(1049) 565	1050 (566)	1050 (566)	1025 (552)	1000 (538)
2nd reheat temp.	^{o}F (^{o}C)	-	-	-	-	-	-	-	-	-	1050 (566)	1050 (566)	1000 (538)
Final feed temp.	^{o}F (^{o}C)	-	175 (79)	340 (171)	385 (196)	435 (224)	460 (238)	485 (252)	(485) 252	505 (263)	547 (286)	548 (287)	554 (290)
No. of feed heaters		-	2	3	4	6	6	7	8	8	9	7	8
Condenser Vacuum (Pressure)	in Hg (KN/m^2)	26 (13.5)	28.5 - 29.0 (5.1 - 3.4)			28.9 (3.7)	28.9 (3.7)	28.7 (4.4)	28.4 (5.4)	28.7 (4.4)	28.5 (5.1)	28.5 (5.1)	28.5 (5.1)
Overall efficiency	%		~17	27.6	30.5	35.6	37.5	39.8	39.5	~40			

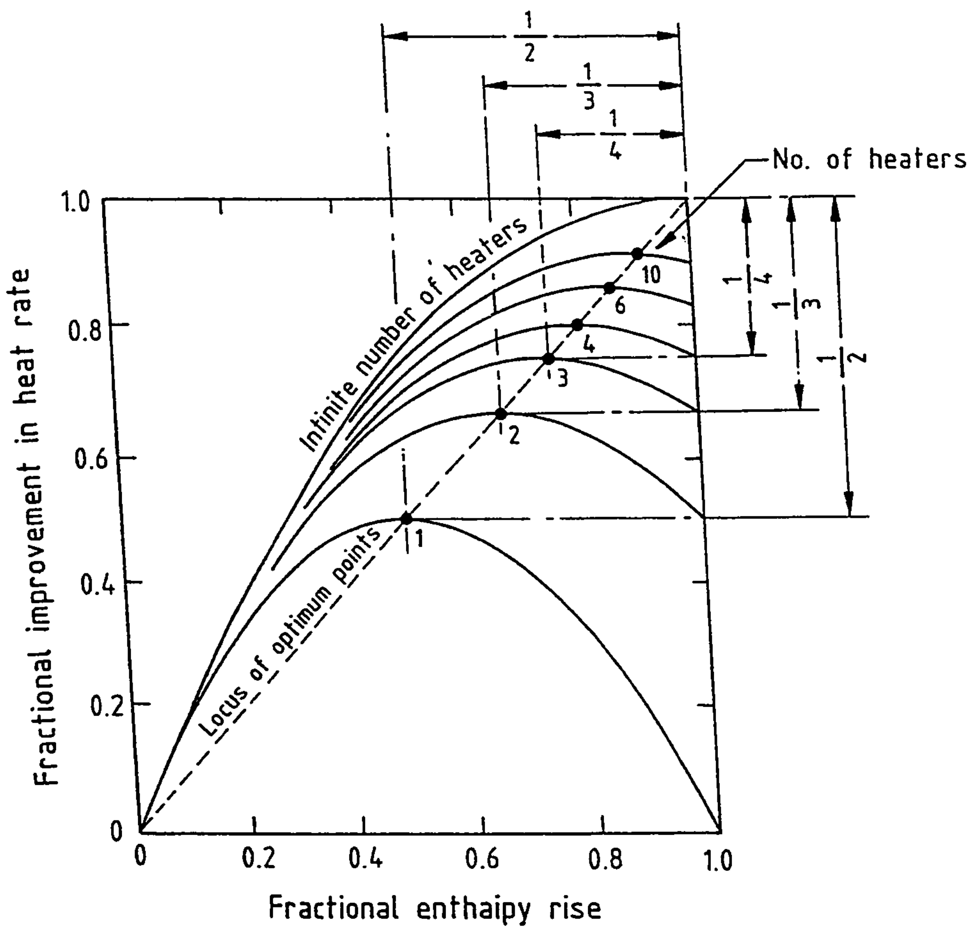


Fig. 4.24 Dimensionless plot of improvement in heat due to feed heating (32)

feedwater heaters increases, but the extra gain due to the addition of a further heater falls. However, the number of feedwater heaters is limited by the number of possible extraction points in the turbines and by econmic considerations.

The effect of number of feedwater heaters on performance varies with initial steam conditions and cycle arrangement. Figures 4.25 and 4.26 illustrate the effect of the extraction pressure ratio (extraction pressure to throttle pressure) on the performance of the existing plant with the addition of 1, 2 and 3 more closed feedwater heaters. The figure indicates that there is an optimum extraction pressure ratio for each feedwater heater added. These ratios are 8.4%, 30% and 40% for heater number 4, 5 and 6 respectively.

To study the effectiveness of adding more feedwater heaters to the existing plant, the cycle arrangement shown in Fig. 4.27 was selected. The results of the first- and second-law analysis are shown in Fig. 4.28. An increase in the number of feedwater heaters from three to six at a throttle pressure of 12.5 MPa improved the first-law efficiency by 3% , and the second-law efficiency by 1%. The efficiencies reported in Fig. 4.28 are the ones calculated at optimum extraction pressure ratios.

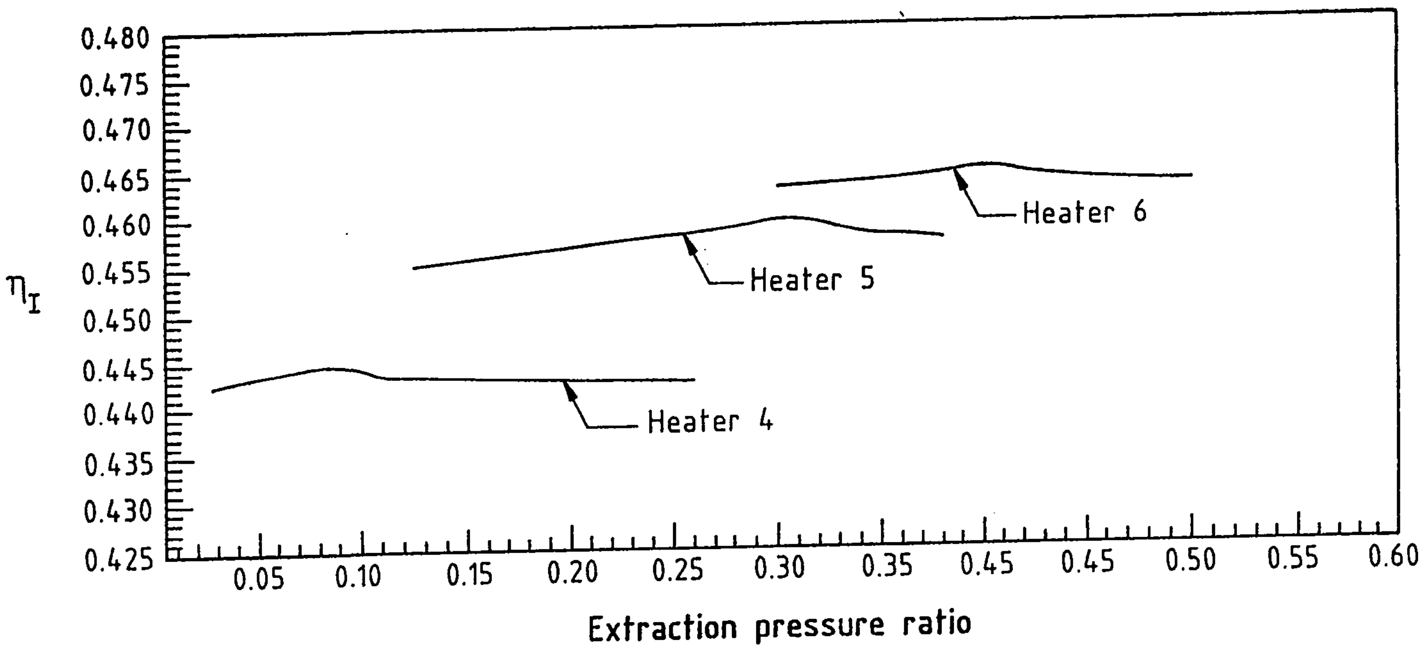


Fig. 4.25 Effect of extracted pressures from turbines to feedwater heaters on the first law efficiency.

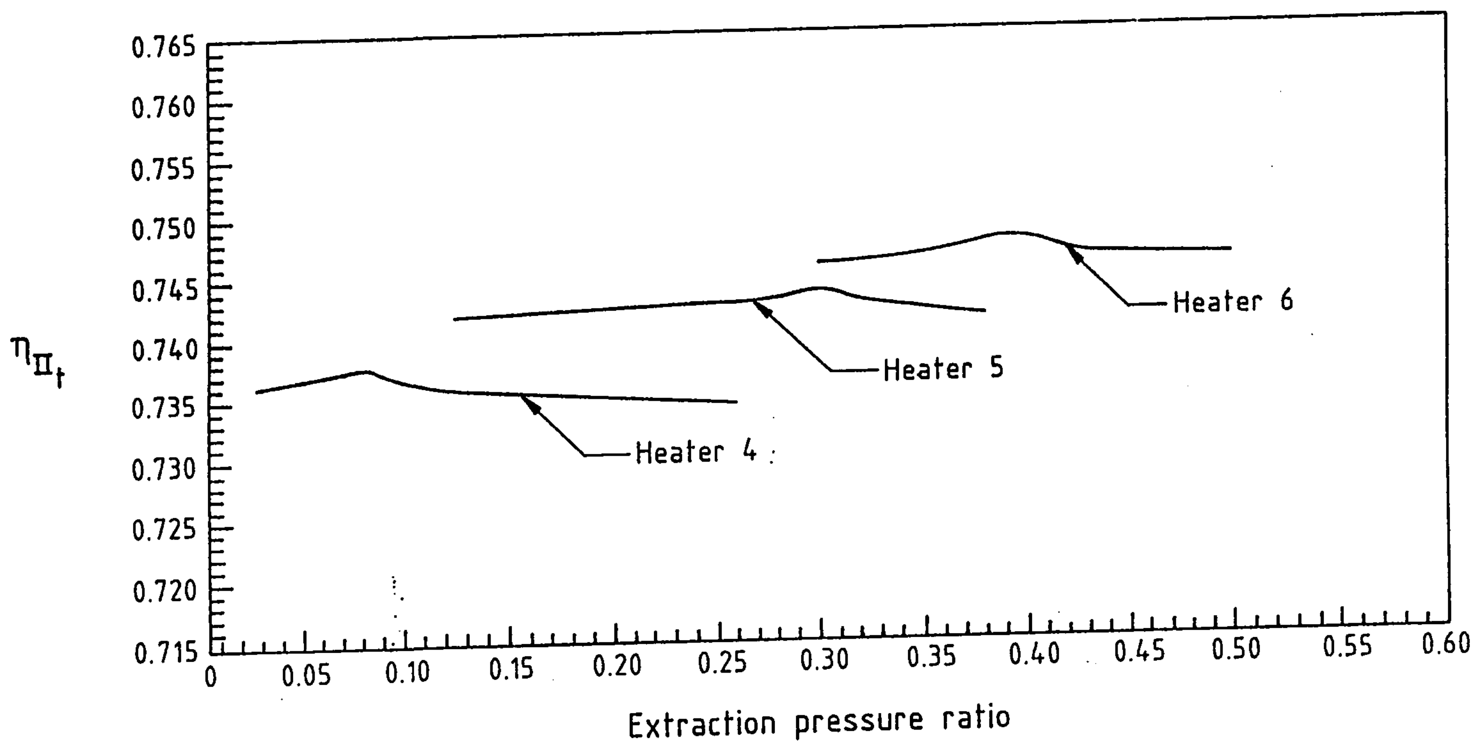


Figure 4.26 Effect of extracted pressure from turbine to feed feedwater heaters on the second law efficiency.

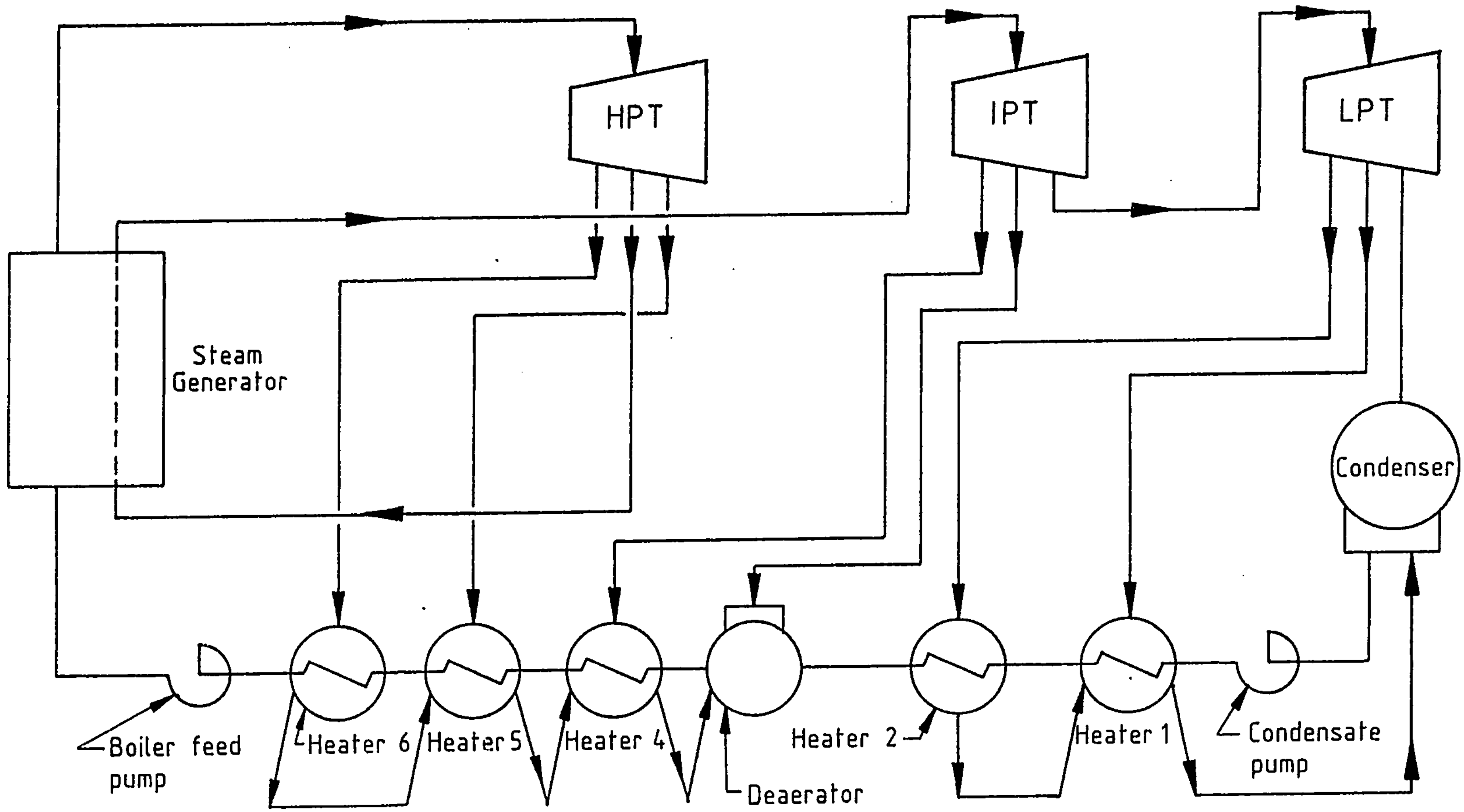


Fig. 4.27 Cycle arrangement for six feedwater heaters.

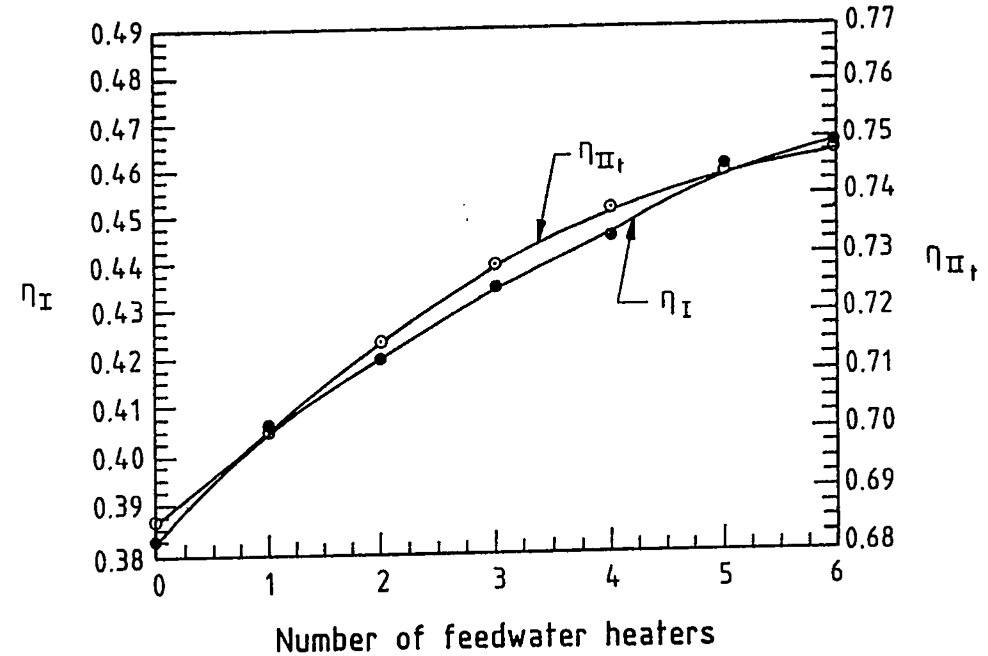


Fig. 4.28 Effect of number of feedwater heaters on the performance at optimum extraction pressure ratios.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

A computer program was written to perform the necessary calculations required to study the actual and design performance of Ghazlan Power Plant. The results indicated that there is a room for improvement in the plant performance. Being encouraged by these results and the fact that improvement in the performance will result in savings of millions of Saudi Riyals per year, a full energy and exergy analysis were carried out to identify the potential for improving the performance of the plant. A broad range of operating conditions and physical quantities, such as throttle steam pressure and temperature, reheat pressure and temperature for a single reheat, reheat pressure for double reheat, and number of feedwater heaters were studied. The results of the parametric study are summarized as follows:

- The addition of three more feedwater heaters increases the turbine cycle efficiency by 3%.
- Increasing the throttle temperature by 10% increases the turbine cycle efficiency by 0.5% .
- The study shows that the optimum reheat pressure for a single reheat is 3.83 MPa, however, the operating reheat pressure is around 2.85 MPa. Operating at the optimum reheat condition will improve the plant efficiency by about 0.05%.

- The study shows that having 2 reheat instead of a single reheat will improve the turbine cycle efficiency by 1.1% if operated at the optimum conditions where the optimum pressures of a double reheat are 4.5% and 24% of the throttle pressure.

- Increasing the throttle pressure from 12 MPa (present value) to 18 MPa will improve the turbine cycle efficiency by 1%.

- Increasing the degree of reheat by 100^{o}K will improve the turbine cycle efficiency by about 0.44%.

5.2 RECOMMENDATIONS FOR FURTHER STUDIES

As an extension for this study, it is recommended that the following points be studied:

(i) A detailed study on the effect of different flame temperature on the performance of the plant.

(ii) An exergetic-economic analysis of the plant and of the different potential options available for the plant improvement.

(iii) Consider the effect of CO, NO formation in the combustion process.

APPENDIX-A

```
****************************************************
*  NOMENCLATURE OF THE PROGRAM
****************************************************
* MF : FUEL FLOW RATE IN KG/S
* E  : EXCESS AIR
* TO : SURROUNDING TEMPERATURE
* ACH: CHEMICAL AVALIBLITIY
* ATH: THERMOMECHANICAL AVALIBILITY
* MBE: STEAM PRODUCED
* TBI: FEEDWATER TEMPERATURE
* PBE: THROTTLE PRESSURE
* TBE: THROTTLE TEMPERATURE
* PRE: REHEAT PRESSURE
* TRE: REHEAT TEMPERATURE
* MH1: EXTRACTED STEAM TO HEATER#1
* MH2: EXTRACTED STEAM TO HEATER#2
* MD : EXTRACTED STEAM TO DERATOR
* PLPTE: EXHAUST PRESSURE
* Q : HEAT INPUT
* QB : HEAT OUTPUT FROM THE BOILER
* WT : WORK OF THE TURBINES
* WTG: WORK OUT OF THE GENERATOR
* IB : IRREVERSIBILITY IN THE BOILER
* PIB: PERCENTAGE OF IRREVERSIBILITY IN THE BOILER TO HEAT INPUT
* ICOMP: IRREVERSIBILITY IN THE COMBUSTION
* PICOMP: PERCENTAGE OF IRREVERSIBILITY IN THE COMBUSTION TO HEAT INPUT
* IT : IRREVERSIBILITY IN THE TURBINES
* PIT: PERCENTAGE OF IRREVERSIBILITY IN THE TURBINES TO HEAT INPUT
* IGEN : IRREVERSIBILITY IN THE GENERATOR
* PIGEN: PERCENTAGE OF IRREVERSIBILITY IN THE GENERATOR TO HEAT INPUT
* QC : HEAT REJECTED IN THE CONDENSER
* IC : IRREVERSIBILITY IN THE CONDENSER
* PIC: PERCENTAGE OF IRREVERSIBILITY IN THE CONDENSER TO HEAT INPUT
* IH : IRREVERSIBILITY IN THE HEATERS
* PIH: PERCENTAGE OF IRREVERSIBILITY IN THE HEATERS TO HEAT INPUT
* ID : IRREVERSIBILITY IN THE DEARATOR
* PID: PERCENTAGE OF IRREVERSIBILITY IN THE DEARATOR TO HEAT INPUT
* EB : EFFICIENCY OF THE BOILER
* ET : EFFICIENCY OF THE TURBINE
* EGEN : EFFICIENCY OF THE GENERATOR
* E1 : FIRST LAW EFFICIENCY
* E2 : SECOND LAW EFFICIENCY
*******************************************************************
C      IMPLICIT DOUBLE PRECISION (A-H,N-Z)
       REAL MF,N2,MW,HHV
C      REAL ICOMP,NR,NP
       REAL IB,IT,MBI,MR,MHPTI,MIPTI,MLPTI,IHPT,IIPT,ILPT
       REAL MBE,MHPTE,MIPTE,MLPTE
       REAL MBL,MHPTL,MIPTL,MLPTL
       REAL MH,IH1,IH2,IH,ICP,ITOT,MD
       REAL MCI1,MCI3,MCE,IC,IGEN
       REAL MDI1,MDI2,MDI3,MDE,ID,MBFP,IBFP
       EXTERNAL CPCO2,CPH2O,CPO2,CPN2
       REAL TBI,PBI,HBI,SBI
       REAL HCO2,HH2O,HO2,HN2
```

```
      REAL HOCO2,HOH2O,HOO2,HON2
*
      COMMON
     1P(10),T(10),EFF(10),PROCM(10),S(10),H(10),
     2BKO(5),AKS(9),NS(8),LK(8),XDF(10),HD(10),
     3ZS(8,3),XS(8,2),BK(9,6),AK(22),AS(12),BS(8,2),
     4TC,PC,VC,TH,BETA,HNORM,SNORM,SR,
     5A0,B0,B90,
     6BSMAL,ALZERO,ALONE,ALTWO,AIONE,
     7THETA1,TEHTA2,THETA3,THETAT,
     8BETA1,BETA2
C
      CALL INIT
************************
*     WRITE(6,19)PC,TC
* 19  FORMAT(2E12.3)
*     BETA=5000./PC
*     WRITE(6,*)TC
*     CALL CALCT
*     CALL CALCHC
*     H(1)=HNORM*PC*VC
*     WRITE(6,*)H(1),PC
*
C     COMPOSITION ON A MOLAR BASIS FOR ENVIRONMENT
      YECO2=.0003
      YEH2O=.0303
      YEO2=.2035
      YEN2=.7567
C     WRITE(6,*) 'DESIGN DATA FOR LOAD FACTOR=0.25'
C     **ENTER THE FLOW RATE OF FUEL**
      MF=21.354
C     WRITE(6,*)'MF=',MF
      CO2=.006
      N2=.01
      CH4=.894
      C2H6=.086
      C3H8=.004
      E=1.15
      MW=44.*CO2+N2*28+CH4*16+C2H6*30+C3H8*44
      B=CO2+CH4+2.*C2H6+3.*C3H8
      C=2.*CH4+3.*C2H6+4.*C3H8
      A=B+.5*C-CO2
      D=N2+A*(E+1)*3.76
C     HHVI=(CH4*890800.+C2H6*1542000.+C3H8*2203000.)/MW
      TF=1600.
      TFG=450.
      HHV=50607.177
      Q=MF*HHV
      T0=298.2
      R=8.314
**********************
      HCO2=15483
      HOCO2=9364
      SCO2=230.194
      SOCO2=213.685
```

```
**********************
      HH2O=15080
      HOH2O=9904
      SH2O=202.734
      SOH2O=188.72
**********************
      HO2=13228
      HOO2=8682
      SO2=217.342
      SOO2=205.033
**********************
      HN2=13105
      HON2=8669
      SN2=203.523
      SON2=191.502
**********************
      PCO2=CO2+CH4+2.*C2H6+3.*C3H8
      PH2O=2.*CH4+3.*C2H6+4.*C3H8
      PO2=(E-1)*(PCO2+.5*PH2O)
      PN2=N2+3.76*E*(PCO2+.5*PH2O)
      PTOT=PCO2+PH2O+PO2+PN2
      YCO2=PCO2/PTOT
      YH2O=PH2O/PTOT
      YO2=PO2/PTOT
      YN2=PN2/PTOT
      ACHCO2=MF*R*TO/MW*(PCO2*LOG(YCO2/YECO2))
      ACHH2O=MF*R*TO/MW*(PH2O*LOG(YH2O/YEH2O))
      ACHO2=MF*R*TO/MW*(PO2*LOG(YO2/YEO2))
      ACHN2=MF*R*TO/MW*(PN2*LOG(YN2/YEN2))
      ATHCO2=MF/MW*PCO2*((HCO2-HOCO2)-TO*(SCO2-SOCO2))
      ATHH2O=MF/MW*PH2O*((HH2O-HOH2O)-TO*(SH2O-SOH2O))
      ATHO2=MF/MW*PO2*((HO2-HOO2)-TO*(SO2-SOO2))
      ATHN2=MF/MW*PN2*((HN2-HON2)-TO*(SN2-SON2))
      SIGMA=26.6299
      AF=Q
      AQ=Q*(1-TO/TF)
      ACH=ACHCO2+ACHH2O+ACHO2+ACHN2
      ATH=ATHCO2+ATHH2O+ATHO2+ATHN2
      WRITE(6,*)'******************************'
      WRITE(6,*)'FUEL COMPSITION'
      WRITE(6,*)'******************************'
      WRITE(6,*)'PERCENTAGE OF CO2 PER ONE MOLE OF FUEL =',CO2
      WRITE(6,*)'PERCENTAGE OF N2 PER ONE MOLE OF FUEL =',N2
      WRITE(6,*)'PERCENTAGE OF CH4 PER ONE MOLE OF FUEL =',CH4
      WRITE(6,*)'PERCENTAGE OF C2H6 PER ONE MOLE OF FUEL =',C2H6
      WRITE(6,*)'PERCENTAGE OF C4H8 PER ONE MOLE OF FUEL =',C4H8
      WRITE(6,*)'******************************'
      WRITE(6,*)'EXCESS AIR =',E-1,'%'
      WRITE(6,*)'THE FLAME TEMPERATURE =',TF,'DEG K'
      WRITE(6,*)'THE FLUE GAS TEMPERATURE =',TFG,'DEG K'
      WRITE(6,*)'THE SURROUNDING TEMPERATURE =',TO-273.2,'DEG C'
      WRITE(6,*)'THE FUEL FLOW RATE =',MF,'KG/S'
      WRITE(6,*)'THE HIGER HEATING VALUE OF THE FUEL =',HHV,'KJ/KG'
      ICOMP=AF-ATH-ACH-AQ
C     WRITE(6,*)'ICOMP',ICOMP
```

```
      PICOMP=ICOMP/Q
C     WRITE(6,*)'PICOMP',PICOMP
**************************
****** INPUT DATA   *******
**************************
      TBI=174.1+273.2
      MBI=323.04
      PBE=8000.
      TBE=510.00+273.2
      MBE=321.4
      MHPTI=MBE
      PHPTE=3166.9
      MHPTE=MHPTI*.95
      MR=318.74
      TRE=510.00+273.2
      PRE=2850.4
      PIPTE=861.9
      PLPTE=5.318
      MH=271.7
      MH1=28.82
      MH2=15.77
      PH112=132.1
      PH212=362.48
**************************
      MBL=MBI-MBE
      MHPTL=MHPTI-MHPTE
      MIPTI=MR
      MIPTE=MR*.97
      MIPTL=MIPTI-MIPTE
      MLPTI=285.2
      MLPTE=227.1
      MD=MIPTE-MLPTI
      MDI1=MH
      MDI2=0.5*MD
      MDI3=0.5*MD
      MDE=MDI1+MDI2+MDI3
**************************
****** BOILER INLET    ****
**************************
      TH=TBI/TC
      CALL PSAT(BETAK)
      PBI=BETAK*PC
      BETA=PBI/PC
      CALL CALCHC
      HBI=HNORM*PC*VC
      CALL CALCSC
      SBI=SNORM*PC*VC/TC
C     WRITE(6,*)'HBI=',HBI,'SBI=',SBI
      WRITE(6,*)'**************************'
      WRITE(6,*)'THE FEEDWATER TEMPERATUR =',TBI,'DEG K'
      WRITE(6,*)'THROTTLE TEMPERATURE =',TBE,'DEG K'
      WRITE(6,*)'THROTTLE PRESSURE =',PBE,'KPA'
      WRITE(6,*)'STEAM PRODUCED =',MBE,'KG/S'
      WRITE(6,*)'REHEAT TEMPERATURE =',TRE,'DEG K'
      WRITE(6,*)'REHEAT PRESSURE =',PRE,'KPA'
```

```
      WRITE(6,*)'*****************************'
**************************
****** BOILER EXIT    ****
**************************
      BETA=PBE/PC
      TH=TBE/TC
      CALL CALCHH
      HBE=HNORM*PC*VC
      CALL CALCSH
      SBE=SNORM*PC*VC/TC
C     WRITE(6,*)'HBE=',HBE,'SBE=',SBE
**************************
**************************
      HBL=(HBI+HBE)/2.
      PL=101.
      BETA=PL/PC
      HBLN=HBL/(PC*VC)
      CALL CALCS(HBLN,SN)
      SBL=SN*PC*VC/TC
C     WRITE(6,*)'HBL=',HBL,'SBL=',SBL
**************************
******H.TURBINE INLET  ****
**************************
      HHPTI=HBE
      SHPTI=SBE
C     WRITE(6,*)'HHPTI=',HHPTI,'SHPTI=',SHPTI
**************************
******H.TURBINE EXIT ******
**************************
      EFFT=.766
      SHPTSN=SHPTI/(PC*VC/TC)
      BETA=PHPTE/PC
      CALL CALCH(SHPTSN,HHPTSN)
      HHPTES=HHPTSN*(PC*VC)
      HHPTE=HHPTI-EFFT*(HHPTI-HHPTES)
      HHPTEN=HHPTE/(PC*VC)
      CALL CALCS(HHPTEN,SHPTEN)
      SHPTE=SHPTEN*(PC*VC/TC)
C     WRITE(6,*)'HHPTE=',HHPTE,'SHPTE=',SHPTE
**************************
**************************
      HHPTL=(HHPTI+HHPTE)/2.
      PL=101.
      BETA=PL/PC
      HHPTLN=HHPTL/(PC*VC)
      CALL CALCS(HHPTLN,SN)
      SHPTL=SN*PC*VC/TC
C     WRITE(6,*)'HHPTL=',HHPTL,'SHPTL=',SHPTL
**************************
******REHEAT INLET  ****
**************************
      HRI=HHPTE
      SRI=SHPTE
C     WRITE(6,*)'HRI=',HRI,'SRI=',SRI
**************************
```

```
******REHEAT EXIT ******
**************************
      BETA=PRE/PC
      TH=TRE/TC
      CALL CALCHH
      HRE=HNORM*PC*VC
      CALL CALCSH
      SRE=SNORM*PC*VC/TC
C     WRITE(6,*)'HRE=',HRE,'SRE=',SRE
**************************
******I.TURBINE INLET  ****
**************************
      HIPTI=HRE
      SIPTI=SRE
C     WRITE(6,*)'HIPTI=',HIPTI,'SIPTI=',SIPTI
**************************
******I.TURBINE EXIT ******
**************************
      SIPTSN=SIPTI/(PC*VC/TC)
      BETA=PIPTE/PC
      CALL CALCH(SIPTSN,HIPTSN)
      HIPTES=HIPTSN*(PC*VC)
      HIPTE=HIPTI-EFFT*(HIPTI-HIPTES)
      HIPTEN=HIPTE/(PC*VC)
      CALL CALCS(HIPTEN,SIPTEN)
      SIPTE=SIPTEN*(PC*VC/TC)
C     WRITE(6,*)'HIPTE=',HIPTE,'SIPTE',SIPTE
**************************
**************************
      HIPTL=(HIPTI+HIPTE)/2.
      PL=101.
      BETA=PL/PC
      HIPTLN=HIPTL/(PC*VC)
      CALL CALCS(HIPTLN,SN)
      SIPTL=SN*PC*VC/TC
C     WRITE(6,*)'HIPTL=',HIPTL,'SIPTL=',SIPTL
****************************
******L.TURBINE INLET  ****
**************************
      HLPTI=HIPTE
      SLPTI=SIPTE
C     WRITE(6,*)'HLPTI=',HLPTI,'SLPTI=',SLPTI
**************************
******L.TURBINE EXIT ******
**************************
      SLPTSN=SLPTI/(PC*VC/TC)
      BETA=PLPTE/PC
      CALL CALCH(SLPTSN,HLPTSN)
      HLPTES=HLPTSN*(PC*VC)
      HLPTE=HLPTI-EFFT*(HLPTI-HLPTES)
      HLPTEN=HLPTE/(PC*VC)
      CALL CALCS(HLPTEN,SLPTEN)
      SLPTE=SLPTEN*(PC*VC/TC)
C     WRITE(6,*)'HLPTE=',HLPTE,'SLPTE=',SLPTE
**************************
```

```
C*****L.TURBINE EXIT ******
C*****  FOR HEATER 1 *****
C************************
      SLPTSN=SLPTI/(PC*VC/TC)
      BETA=PH112/PC
      CALL CALCH(SLPTSN,HN)
      HH1S=HN*(PC*VC)
      HH112=HLPTI-EFFT*(HLPTI-HH1S)
      HH1N=HH112/(PC*VC)
      CALL CALCS(HH1N,SH1N)
      SH112=SH1N*(PC*VC/TC)
C     WRITE(6,*)'HH112=',HH112,'SH112=',SH112
C************************
C*****L.TURBINE EXIT ******
C*****  FOR HEATER 2 *****
C************************
      SLPTSN=SLPTI/(PC*VC/TC)
      BETA=PH212/PC
      CALL CALCH(SLPTSN,HN)
      HH2S=HN*(PC*VC)
      HH212=HLPTI-EFFT*(HLPTI-HH2S)
      HH2N=HH212/(PC*VC)
      CALL CALCS(HH2N,SH2N)
      SH212=SH1N*(PC*VC/TC)
C     WRITE(6,*)'HH212=',HH212,'SH212=',SH212
C************************
C************************
      HLPTL=(HLPTI+HLPTE)/2.
      PL=101.
      BETA=PL/PC
      HLPTLN=HLPTL/(PC*VC)
      CALL CALCS(HLPTLN,SN)
      SLPTL=SN*PC*VC/TC
C     WRITE(6,*)'HLPTL=',HLPTL,'SLPTL=',SLPTL
C*********************************
C*****CONDENCER INLET#1 (MAIN) ****
C*********************************
      HCI1=HLPTE
      SCI1=SLPTE
C     WRITE(6,*)'HCI1=',HCI1,'SCI1=',SCI1
C*********************************
C*****CONDENCER EXIT  (MAIN) *******
C*********************************
      TCE=38.38+273.2
      TH=TCE/TC
      CALL PSAT(BETA)
      CALL CALCHC
      HCE=HNORM*PC*VC
      CALL CALCSC
      SCE=SNORM*PC*VC/TC
C     WRITE(6,*)'HCE=',HCE,'SCE=',SCE
C***********************************
C*****BOILER FEED PUMP TURBINE INLET**
C***********************************
      PBFPTI=359.232
```

```
      BETA=PBFPTI/PC
      SBFPTI=SLPTI
      SN=SBFPTI/(PC*VC/TC)
      CALL CALCH(SN,HN)
      HNS=HN*(PC*VC)
      HBFPTI=HLPTI-EFFT*(HLPTI-HNS)
      HN=HBFPTI/(PC*VC)
      CALL CALCH(HN,SN)
      SBFPTI=SN*(PC*VC/TC)
C     WRITE(6,*)'HBFPTI=',HBFPTI,'SBFPTI=',SBFPTI
C***************************************
C*****BOILER FEED PUMP TURBINE EXIT**
C***************************************
      PBFPTE=350.
      EFFT=.816
      BETA=PBFPTE/PC
      SBFPTS=SBFPTI
      SN=SBFPTS/(PC*VC/TC)
      CALL CALCH(SN,HN)
      HNS=HN*(PC*VC)
      HBFPTE=HBFPTI-EFFT*(HBFPTI-HNS)
      HN=HBFPTE/(PC*VC)
      CALL CALCH(HN,SN)
      SBFPTE=SN*(PC*VC/TC)
C     WRITE(6,*)'HBFPTE=',HBFPTE,'SBFPTE=',SBFPTE
C***************************************
C*****INLET CONDENSATE PUMP   ********
C***************************************
        PCPI=PCE
        HCPI=HCE
        SCPI=SCE
C     WRITE(6,*)'HCPI=',HCPI,'SCPI=',SCPI
C***************************************
C*****EXIT CONDENSATE PUMP  **********
C***************************************
      EFFP=.735
      TCPE=38.83+273.2
      TH=TCPE/TC
      CALL PSAT(BETA)
      CALL CALCHC
      HCPE=HNORM*(PC*VC)
      CALL CALCSC
      SCPE=SNORM*(PC*VC/TC)
C     WRITE(6,*)'HCPE=',HCPE,'SCPE=',SCPE
C***************************************
C***** INLET #1 HETAR #1      ********
C***************************************
      TH111=39.167+273.2
      TH=TH111/TC
      CALL PSAT(BETA)
      PH111N=BETA
      CALL CALCHC
      HH111=HNORM*(PC*VC)
      CALL CALCSC
      SH111=SNORM*(PC*VC/TC)
```

```
C     WRITE(6,*)'HH111=',HH111,'SH111=',SH111
*************************************
****** INLET #2 HETAR #1      ********
*************************************
      SH112=SLPTE
      SN=SH112/(PC*VC/TC)
      BETA=PH112/PC
      CALL CALCH(SN,HN)
      HH112=HN*(PC*VC)
C     WRITE(6,*)'HH112=',HH112,'SH112=',SH112
*************************************
******INLET #3 HETAR #1      **********
*************************************
      TH113=91.611+273.2
      TH=TH113/TC
      CALL PSAT(BETA)
      CALL CALCHC
      HH113=HNORM*(PC*VC)
      CALL CALCSC
      SH113=SNORM*(PC*VC/TC)
C     WRITE(6,*)'HH113=',HH113,'SH113=',SH113
**************************
*************************************
******EXIT #2 HETAR #2      **********
*************************************
      HH2E2=HH113
      SH2E2=SH113
C     WRITE(6,*)'HH2E2=',HH2E2,'SH2E2=',SH2E2
*************************************
******EXIT #2 HETAR #1      **********
*************************************
      TH1E2=44.89+273.2
      TH=TH1E2/TC
      CALL PSAT(BETA)
      CALL CALCHC
      HH1E2=HNORM*(PC*VC)
      CALL CALCSC
      SH1E2=SNORM*(PC*VC/TC)
C     WRITE(6,*)'HH1E2=',HH1E2,'SH1E2=',SH1E2
*************************************
******EXIT #1 HETAR #1      **********
*************************************
      HH1E1=(MH1*HH112+MH2*HH113-(MH1+MH2)*HH1E2)/MH+HH111
C     TH1E1=(HH1E1/4.184)+273.2
      TH1E1=109.+273.2
      TH=TH1E1/TC
      CALL PSAT(BETA)
      CALL CALCSC
      SH1E1=SNORM*(PC*VC/TC)
C     WRITE(6,*)'HH1E1=',HH1E1,'SH1E1=',SH1E1
C     WRITE(6,*)'TH1E1=',TH1E1
*************************************
******CONDENSER INLET#3      **********
*************************************
      HC13=HH1E2
```

```
      SCI3=SH1E2
C     WRITE(6,*)'HCI3=',HCI3,'SCI3=',SCI3
**************************************
****** HETAR #2 INLET#1     **********
**************************************
      HH2I1=HH1E1
      SH2I1=SH1E1
C     WRITE(6,*)'HH2I1=',HH2I1,'SH2I1=',SH2I1
**************************************
******INLET#2 HETAR #2      **********
**************************************
      SH2I2=SLPTE
      SN=SH2I2/(PC*VC/TC)
      BETA=PH2I2/PC
      CALL CALCH(SN,HN)
      HH2I2=HN*(PC*VC)
C     WRITE(6,*)'HH2I2=',HH2I2,'SH2I2=',SH2I2
**************************************
******EXIT #1 HETAR #2      **********
**************************************
      HH2E1=MH2*(HH2I2-HH2E2)/MH+HH2I1
C     TH2E1=(HH2E1/4.184)+273.2
      TH2E1=110.+273.2
      TH=TH2E1/TC
      CALL PSAT(BETA)
      CALL CALCSC
      SH2E1=SNORM*(PC*VC/TC)
C     WRITE(6,*)'HH2E1=',HH2E1,'SH2E1=',SH2E1
C     WRITE(6,*)'TH2E1=',TH2E1
**************************************
******INLET#1 DEAERATOR     **********
**************************************
      TDI1=TH2E1
      TH=TDI1/TC
      CALL PSAT(BETA)
      CALL CALCHC
      HDI1=HNORM*PC*VC
      CALL CALCSC
      SDI1=SNORM*(PC*VC/TC)
C     WRITE(6,*)'HDI1=',HDI1,'SDI1=',SDI1
**************************************
******INLET#2 DEAERATOR     **********
**************************************
      PDI2=860.37
      BETA=PDI2/PC
      SDI2=SIPTE
      SN=SDI2/(PC*VC/TC)
      CALL CALCH(SN,HN)
      HDI2=HN*PC*VC
C     WRITE(6,*)'HDI2=',HDI2,'SDI2=',SDI2
**************************************
******INLET#3 DEAERATOR     **********
**************************************
      TDI3=157.05+273.2
      TH=TDI3/TC
```

```
      CALL PSAT(BETA)
      CALL CALCHC
      HDI3=HNORM*PC*VC
      CALL CALCSC
      SDI3=SNORM*(PC*VC/TC)
C     WRITE(6,*)'HDI3=',HDI3,'SDI3=',SDI3
C*************************************
C***** EXIT DEAERATOR     **********
C*************************************
      HDE=(MDI1*HDI1+MDI2*HDI2+MDI3*HDI3)/MDE
C     TDE=(HDE/4.184)+273.2
      TDE=149.+273.2
      TH=TDE/TC
      CALL PSAT(BETA)
      CALL CALCSC
      SDE=SNORM*(PC*VC/TC)
C     WRITE(6,*)'HDE=',HDE,'SDE=',SDE
C     WRITE(6,*)'TDE=',TDE
C*************************************
C*****BOILER FEED PUMB INLET *********
C*************************************
      HBFPI=HDE
      SBFPI=SDE
C     WRITE(6,*)'HBFPI=',HBFPI,'SBFPI=',SBFPI
C*************************************
C*****BOILER FEED PUMB EXIT *********
C*************************************
      HBFPE=HBFPI+11.
C     TBFPE=(HBFPE/4.184)+273.2
      TBFPE=174.+273.2
      TH=TBFPE/TC
      CALL PSAT(BETA)
      CALL CALCSC
      SBFPE=SNORM*(PC*VC/TC)
C     WRITE(6,*)'HBFPE=',HBFPE,'SBFPE=',SBFPE
C     WRITE(6,*)'TBFPE=',TBFPE
C**************************
C*** BOILER INLET MODIFIED**
C**************************
C     WRITE(6,*)'TBI=',TBI
      HBI=HBFPE
      SBI=SBFPE
      TBI=TBFPE
C     WRITE(6,*)'HBI=',HBI,'SBI=',SBI
C     WRITE(6,*)'TBI=',TBI
C**************************
      T0=298.2
      QB=MBE*HBE-MBI*HBI+MR*(HRE-HRI)+MBL*HBL
      Q=QB/.8637
      MF=Q/HHV
      EXB1=MBE*(HBE-T0*SBE)-MBI*(HBI-T0*SBI)
      EXB2=MBL*(HBL-T0*SBL)+MR*((HRE-HRI)-T0*(SRE-SRI))
      EXB=EXB1+EXB2
      IB=T0*(MBE*SBE-MBI*SBI+MR*(SRE-SRI)-QB/TF+MBL*SBL)
      PIB=IB/Q
```

```
      EB=QB/Q
C     WRITE(6,*)'EXB=',EXB
C     WRITE(6,*)'QB=',QB
C     WRITE(6,*)'EB=',EB
C     WRITE(6,*)'Q=',Q
C     WRITE(6,*)'MF=',MF
C     WRITE(6,*)'QB=',QB,'IB=',IB,'EB=',EB
      WRITE(6,*)'PIB=',PIB
*******
      WHPT=MHPTE*(HHPTI-HHPTE)
      WIPT=MIPTE*(HIPTI-HIPTE)
      WLPT=MLPTI*HLPTI-MLPTE*HLPTE-MH1*HH1I2-MH2*HH2I2
      IHPT=TO*(MHPTE*SHPTE-MHPTI*SHPTI+MHPTL*SHPTL)
      IIPT=TO*(MIPTE*SIPTE-MIPTI*SIPTI+MIPTL*SIPTL)
      ILPT=TO*(MLPTE*SLPTE-MLPTI*SLPTI+MH1*SH1I2+MH2*SH2I2)
      IT=IHPT+IIPT+ILPT
      WT=WHPT+WIPT+WLPT
      EGEN=0.985
      IGEN=WT-EGEN*WT
      WTG=WT-IGEN
      ET=WT/(WT+IT)
      PIT=IT/Q
      PIHPT=IHPT/Q
      PIIPT=IIPT/Q
      PILPT=ILPT/Q
C     WRITE(6,*)'WT=',WT,'IT=',IT,'ET=',ET
C     WRITE(6,*)'IHPT=',IHPT,'IIPT=',IIPT,'ILPT=',ILPT
C     WRITE(6,*)'PIHPT=',PIHPT,'PIIPT=',PIIPT,'PILPT=',PILPT
C     WRITE(6,*)'PIT=',PIT
*******
      MCI1=MH-MH1-MH2
      MCI3=MH1+MH2
      MCE=MH
      QC=MCI1*HCI1+MCI3*HCI3-MCE*HCE
      IC=TO*(MCE*SCE-MCI1*SCI1-MCI3*SCI3)+QC
      PIC=IC/Q
      EC=QC/(QC+IC)
C     WRITE(6,*)'QC=',QC,'IC=',IC,'EC=',EC
C     WRITE(6,*)'PIC=',PIC
*******
      QH1=MH*(HH1E1-HH1I1)
      IH1=TO*(MH*(SH1E1-SH1I1)+(MH1+MH2)*SH1E2-MH1*SH1I2-MH2*SH1I3)
C     WRITE(6,*)'QH1=',QH1,'IH1=',IH1
*******
      QH2=MH*(HH2E1-HH2I1)
      IH2=TO*(MH*(SH2E1-SH2I1)-MH2*(SH2E2-SH2I2))
      WRITE(6,*)'QH2=',QH2,'IH2=',IH2
      IH=IH1+IH2
      PIH=IH/Q
C     WRITE(6,*)'SH2E1=',SH2E1
C     WRITE(6,*)'SH2I1=',SH2I1
C     WRITE(6,*)'SH2E2=',SH2E2
C     WRITE(6,*)'SH2I2=',SH2I2
C     WRITE(6,*)'IH=',IH
C     WRITE(6,*)'PIH=',PIH
```

```
*******
      WCP=MH*(HCPE-HCPI)
      ICP=MH*TO*(SCPE-SCPI)
C     WRITE(6,*)'WCP=',WCP,'ICP=',ICP
*******
      ID=TO*(MDE*SDE-MDI1*SDI1-MDI2*SDI2-MDI3*SDI3)
C     WRITE(6,*)'ID=',ID
*******
      MBFP=MDE
      WBFP=MBFP*(HBFPE-HBFPI)
      IBFP=TO*MBFP*(SBFPE-SBFPI)
      PIP=(IBFP+ICP)/Q
C     WRITE(6,*)'WBFP=',WBFP,'IBFP=',IBFP
C     WRITE(6,*)'PIP=',PIP
      IGEN=WT-EGEN*WT
      PIGEN=IGEN/Q
C     WRITE(6,*)'IGEN=',IGEN
C     WRITE(6,*)'PIGEN=',PIGEN
      ITOT=(ICOMP+IB+IT+IC+ICP+IH+ID+IBFP+IGEN)
C     WRITE(6,*)'ITOT=',ITOT
      PITOT=ITOT/Q
C     WRITE(6,*)'PITOT=',PITOT
      E1=(WTG-WCP-WBFP)/QB
      E2=(WTG-WCP-WBFP)/EXB
C     WRITE(6,*)'E1=',E1
C     WRITE(6,*)'E2=',E2
      WRITE(6,*)'**************************'
      WRITE(6,*)'HEAT INPUT =',Q,'KW'
      WRITE(6,*)' '
      WRITE(6,*)'IRREVERSIBILITY IN THE COMBUSTION =',ICOMP,'KW'
      WRITE(6,*)'PERCENTAGE IRREVERSIBILITY OF COMBUSTION =',PICOMP
      WRITE(6,*)' '
      WRITE(6,*)'IRREVERSIBILITY IN THE HEAT EXCHANGER=',IB,'KW'
      WRITE(6,*)'PERCENTAGE IRREVERSIBILITY OF HEAT EXCHANGER =',PIB
      WRITE(6,*)'BOILER EFFICIENCY =',EB
      WRITE(6,*)' '
      WRITE(6,*)'WORK PRODUCED BY TURBINES =',WT,'KW'
      WRITE(6,*)'IRREVERSIBILITY IN THE TURBINES =',IT,'KW'
      WRITE(6,*)'PERCENTAGE IRREVERSIBILITY OF TURBINES =',PIT
      WRITE(6,*)'TURBINE EFFICIENCY =',ET
      WRITE(6,*)'IRREVERSIBILITY IN HPT =',IHPT,'KW'
      WRITE(6,*)'PERCENTAGE IRREVERSIBILITY OF HPT =',IHPT/Q
      WRITE(6,*)'IRREVERSIBILITY IN IPT =',IHPT,'KW'
      WRITE(6,*)'PERCENTAGE IRREVERSIBILITY OF IPT =',IIPT/Q
      WRITE(6,*)'IRREVERSIBILITY IN LPT =',LHPT,'KW'
      WRITE(6,*)'PERCENTAGE IRREVERSIBILITY OF LPT =',ILPT/Q
      WRITE(6,*)' '
      WRITE(6,*)'HEAT REJECTED IN THE CONDENSER =',QC,'KW'
      WRITE(6,*)'IRREVERSIBILITY IN THE CONDENSER =',IC
      WRITE(6,*)'PERCENTAGE IRREVERSIBILITY OF CONDENSER =',PIC
      WRITE(6,*)' '
      WRITE(6,*)'IRREVERSIBILITY IN THE HEATERS =',IH
      WRITE(6,*)'PERCENTAGE IRREVERSIBILITY OF HEATERS =',PIH
      WRITE(6,*)' '
      WRITE(6,*)'IRREVERSIBILITY IN THE GENERATOR =',IGEN
```

```
      WRITE(6,*)'PERCENTAGE IRREVERSIBILITY OF GENERATOR =',PIGEN
      WRITE(6,*)'**************************'
      WRITE(6,*)'THERMAL EFFICIENCY OF THE PLANT =',E1
      WRITE(6,*)'FIRST LAW EFFICIENCY OF THE TURBINE CYCLE =',E1/EB
      WRITE(6,*)'SECOND LAW EFFICIENCY OF THE TURBINE CYCLE =',E2
      WRITE(6,*)'**************************'
****************************
      STOP
      END
****************************
      FUNCTION CPCO2(T)
      CPCO2=(9.085+.0024*T-2.77*T*T/10000000)*4.186
      RETURN
      END
      FUNCTION CPH2O(T)
      CPH2O=(8.361+.000492*T+4.46*T*T/10000000)*4.186
      RETURN
      END
      FUNCTION CPO2(T)
      CPO2=(6.935+.000338*T+.43*T*T/10000000)*4.186
      RETURN
      END
      FUNCTION CPN2(T)
      CPN2=(6.935+.000338*T+.43*T*T/10000000)*4.186
      RETURN
      END
```

APPENDIX-B

```
*******************************
FUEL COMPSITION
*******************************
PERCENTAGE OF CO2 PER ONE MOLE OF FUEL = 0.600000098E-02
PERCENTAGE OF N2 PER ONE MOLE OF FUEL = 0.100000016E-01
PERCENTAGE OF CH4 PER ONE MOLE OF FUEL = 0.893999994
PERCENTAGE OF C2H6 PER ONE MOLE OF FUEL = 0.860000253E-01
PERCENTAGE OF C4H8 PER ONE MOLE OF FUEL = 0.000000000E+00
*******************************
EXCESS AIR = 15  %
THE FLAME TEMPERATURE =  1600.00000     DEG K
THE FLUE GAS TEMPERATURE =  450.000000     DEG K
THE SURROUNDING TEMPERATURE =  25.0000000     DEG C
THE FUEL FLOW RATE =  21.3540039     KG/S
THE HIGER HEATING VALUE OF THE FUEL =  50607.1758     KJ/KG
***************************
THE FEEDWATER TEMPERATUR =  447.299805     DEG K
THROTTLE TEMPERATURE =  783.199951     DEG K
THROTTLE PRESSURE =  8000.00000     KPA
STEAM PRODUCED =  321.399902     KG/S
REHEAT TEMPERATURE =  783.199951     DEG K
REHEAT PRESSURE =  2850.39990     KPA
***************************
PIB= 0.262381732
QH2=  45326.3945     IH2=  31909.0508
*************************
HEAT INPUT =  1158521.00     KW

IRREVERSIBILITY IN THE COMBUSTION =      161811 KW
PERCENTAGE IRREVERSIBILITY OF COMBUSTION = 0.149732769

IRREVERSIBILITY IN THE HEAT EXCHANGER=  303974.750     KW
PERCENTAGE IRREVERSIBILITY OF HEAT EXCHANGER = 0.262381732
BOILER EFFICIENCY = 0.863700330

WORK PRODUCED BY TURBINES =  362746.812     KW
IRREVERSIBILITY IN THE TURBINES =  58049.6836     KW
PERCENTAGE IRREVERSIBILITY OF TURBINES = 0.501067154E-01
TURBINE EFFICIENCY = 0.862048149
IRREVERSIBILITY IN HPT =  18143.7969     KW
PERCENTAGE IRREVERSIBILITY OF HPT = 0.156611688E-01
IRREVERSIBILITY IN IPT =  18143.7969     KW
PERCENTAGE IRREVERSIBILITY OF IPT = 0.144864060E-01
IRREVERSIBILITY IN LPT =           0 KW
PERCENTAGE IRREVERSIBILITY OF LPT = 0.199591331E-01

HEAT REJECTED IN THE CONDENSER =  533658.250     KW
IRREVERSIBILITY IN THE CONDENSER =  15651.1875
PERCENTAGE IRREVERSIBILITY OF CONDENSER = 0.135096274E-01

IRREVERSIBILITY IN THE HEATERS =  34682.3867
PERCENTAGE IRREVERSIBILITY OF HEATERS = 0.299367756E-01

IRREVERSIBILITY IN THE GENERATOR =  5441.25000
PERCENTAGE IRREVERSIBILITY OF GENERATOR = 0.469671935E-02
```

```
*************************
THERMAL EFFICIENCY OF THE PLANT = 0.353317857
FIRST LAW EFFICIENCY OF THE TURBINE CYCLE = 0.409074545
SECOND LAW EFFICIENCY OF THE TURBINE CYCLE = 0.693001151
*************************
```

NOMENCLATURE

a	:	specific flow exergy
$\bar{a}_t$	:	total flow exergy per mole of a given flow stream
$\bar{a}_{ti}$	:	specific flow exergy of stream i
A	:	flow availability (exergy)
A_{CH}	:	chemical availability
A_f	:	combustion chamber input availability
A_Q	:	combustion chamber output availability
A_{TM}	:	thermo-mechanical availability
C	:	fuel price (SR/kg)
C_p	:	plant capaity factor
c	:	the molar percentage of number of mole of CO_2 per ne mole of fuel
d	:	the molar percentage of number of N_2 per one mole of fuel
E	:	total energy
f_l	:	the molar percentage of number of mole of C_{nl} H_{ml} per one mole of fuel
HHV	:	fuel higher heating value
h_o	:	the enthalpy when the system is in equilibrium with the surrounding

$\dot{I}$	:	irreversibility rate
$\dot{I}_b$	:	irreversibility rate for the heat exchnger
$\dot{I}_{c.c}$	:	irreversibility rate for the combustion chamber
$\dot{I}_{cond}$	:	irreversibility rate of the condenser
$\dot{I}_{F.H}$	:	feedwater heater irreversibility rate
$\dot{I}_G$	:	generator irreversibility rate
$\dot{I}_p$	:	pump irreversibility rate
$\dot{I}_t$	:	irreversibility rate of the turbine
$\dot{m}_e$	:	steam mass flow rate exit
$\dot{m}_i$	:	steam mass flow inlet
$\dot{m}_f$	:	fuel mass flow rate
M	:	molecular weight of fuel
N	:	number of moles
N_e	:	rate of number of moles of products
N_i	:	rate of number of moles of reactants
N_p	:	production rate of number of moles
P	:	output power

P.L	:	percent load
Q_B	:	heat taken from the boiler
Q_c	:	heat rejected in the condenser
$Q_{c.v}$	:	control volume heat
Q_{in}	:	heat input
$\bar{R}$	:	universal gas constant
r	:	ratio of reheat # 1 pressure to throttle pressure
S_o	:	the entropy when the system is in equilibrium with the surrounding
T_j	:	instantaneous temperature on the boundary
T_o	:	dead-state temperature
W_p	:	work of the pump
$W_{c.v}$	:	Control volume work
W_{rev}	:	reversible work of the turbine
W_t	:	work of the turbine
X	:	mole fraction
X CO_2	:	number of moles of CO_2 in the product per unit mole of fuel
X H_2O	:	number of moles of H_2O in the product per unit mole of fuel

$X\ N_2$	:	number of moles of N_2 in the product per unit mole of fuel
$X\ O_2$	:	number of moles of O_2 in the product per unit mole of fuel
X_{tot}	:	total number of moles of the product per unit mole of fuel
$Y\ CO_2$	:	mole fraction of CO_2 in the product
$Y\ H_2O$	:	mole fraction of H_2O in the product
$Y\ N_2$	:	mole fraction of N_2 in the product
Y_i	:	mole fraction of i in the product
$Y\ O_2$	:	mole fraction of O_2 in the product
$Y^e_{CO_2}$	:	mole fraction of CO_2 in the environment
$Y^e_{H_2O}$	:	mole fraction of H_2O in the environment
Y^e_i	:	mole fraction of i in the environment
$Y^e_{N_2}$	:	mole fraction of N_2 in the environment
$Y^e_{O_2}$	:	mole fraction of O_2 in the environment

GREEK SYMBOLS

$(\alpha\text{-}1)$	:	the percentage excess air by volume
η_b	:	steam generator thermal efficiency
η_g	:	the efficiency of the generator
η_I	:	first law thermal efficiency of the overall power plant
η_{II_G}	:	second law efficiency of the steam generator unit
η_{II_o}	:	second law efficiency of the overall power plant
η_{II_t}	:	second law efficiency of the turbine cycle
η_o	:	overall plant efficiency
η_t	:	turbine second law efficiency
σ	:	entropy production rate

REFERENCES

1. Aheran, J.E., 1980, The Exergy Method of Energy System Analysis, Wiley, New York.

2. Szargut, J., Morris, D.R., and Steward, F.R, 1988, Exergy Analysis of Thermal Chemical, and Metallurgical Processes, Hemisphere, U.S.A.

3. El-Masri, M.A., "On Thermodynamics of Gas Turbine Cycles," ASME Journal of Engineering for Gas Turbines and Power, Vol. 107, Oct. 1985, pp. 880-889.

4. El-Masri, M.A., "Exergy Analysis of Combined Cycles: Part 1," ASME Journal of Engineering for Gas Turbines and Power, Vol. 109, Apr. 1987, pp. 228-236.

5. El-Masri, M.A., and Chin, W.W., "Exergy Analysis of Combined Cycles: Part 2," ASME Journal of Engineering for Gas Turbines and Power, Vol. 109, Apr.. 1987, pp. 237-243.

6. El-Masri, M.A., "GASCAN - An Interactive Code for Thermal Analysis of Gas Turbine Systems," ASME Journal of Engineering for Gas Turbines and Power, Vol. 110, Apr. 1988, pp. 201-209.

7. Porz, E., "A Computer Method for Thermal Power Cycle Calculation," ASME Journal of Engineering for Gas Turbines and Power, Vol. 113, Apr. 1991, pp. 184-189.

8. Tsatsaronis, G., and Sanae, M., "On-line Thermoeconomic Evaluation of Power Plant Performance," International Power Generation Conference, ASME, New York, N.Y., U.S.A., 1991, pp. 1-8.

9. Mastrullo, R., "A Software Rankine Cycles Exergetic Analysis," Advanced Energy Systems Division ASE, Vol. 27, 1992, pp. 187-191.

10. Von Spakovsky, M.R., and Evans, R.B., "The Design and Performance Optimization of Thermal Systems," ASME Journal of Engineering for Gas Turbines and Power, Vol. 112, Jan. 1990, pp. 86-93.

11. Habib, M.A., "Thermodynamic Analysis of the Performance of Cogeneration Plants," Energy, Vol. 17, No. 5, 1992, pp. 485-491.

12. Wong, K.V., and Niu, Z., "Optimization and Different Loads by Minimization of Irreversibilities," ASME Advanced Energy Systems Division, Vol. 25, 1991, pp. 1-5.

13. Bidini, G., and Stecco,S.S., "A Computer Code Using Exergy for Optimizing Thermal Plants," ASME Journal of Engineering for Gas Turbines and Power, Vol. 113, Jan. 1991, pp. 145-150.

14. Bolland, O., "A Comparative Evaluation of Advanced Combined Cycle Alternatives," ASME Journal of Engineering for Gas Turbines and Power, Vol. 113, Apr. 1991, pp. 190-197.

15. Habib, M.A., and Zubair, S.M., "Thermodynamics of Regenerative Rankine Cycle Power Plants," ASME Advanced Energy System Division, Vol. 19, 1990, pp. 29-34.

16. Habib, M.A., and Zubair, S.M., "Second-Law Based Thermodynamic Analysis of Regenerative-Reheat Rankne Cycle Power Plants," Energy, Vol. 17, No. 3, Mar. 1992, pp. 295-301.

17. Gaggioli, R.A., El-Sayed, Y.M., El-Nashar, A.M., and Kamaluddin,B., "Second Law Efficiency and Costing Analysis of a Combined Power and Desalination Plant," ASME Journal of Energy Resources Technology, Vol. 110, No. 2, Jun. 1988, pp. 114-118.

18. Sciubba, E., and Su, T.M., "Second Law Analysis of the Steam Turbine Power Cycle: A Parametric Study," Computer-Aided Engineering of Energy Systems, ASME Winter Annual Meeting, Vol. 2-3, Dec. 1986, pp. 151-165.

19. Gaggioli, R.a., "Coming Fuel Cells with Fuel-Fired Power Plants for Improved Exergy Efficiency," Energy, Vol. 16, No. 10, 1991, pp. 1259-1274.

20. Van Wylen, G.J., and Sonntag, R.E., 1985, Fundamentals of Classical Thermodynamics, Wiley, New York.

21. Weisman, J., and Eckarvt. L.E., 1985, Modern Power Plant Engineering, Prentice Hall.

22. El-Wakil, M.M., 1984, Power Plant Technology, McGraw-Hill.

23. Kivillin, V.A., Sychev, V.V., and Sheindlin, A.E., 1981, Engineering Thermodynamics, Mir Publishers, Moscow.

24. Will, R.K., 1979, Thermodynamics for Engineering Technologists, Marcel Dekker, New York.

25. Jones, J.B., and Hawkins, G.A., 1960, Engineering Thermodynamics, Wiley, New York.

26. Moran, M.J., and Shapiro, H.N., 1988, Fundamentals of Engineering Thermodynamics, Wiley, New York.

27. Haywood, R.W., 1980, Analysis of Engineering Cycles, Maxwell.

28. Li, K.W., and Priddy, A.P., 1985, Power Plant System Design, Wiley, New York.

29. Habib, M.A., El-Mahalawy, Z.M., Safar, Z.S. and Taha, M.R., 1986, "A Computational Technique for the Evaluation of the Performance of the Thermal Power Plant", Proceedings of the 6th Int. Conference for Mechanical Power Engineering, Cairo, Egypt.

30. Private Communications with Saudi ARAMCO (1994).

31. Private Communications with SCECO (1993).

32. Salisbury, J.K., 1942, "The Steam-Turbine Regenerative Cycle - An Analytical Approach", Transactions of ASME, Vol. 64, pp. 231-245.

VITA

NAME : **JAMIL JARALLAH AWADH AL-BAGAWI**

ADDRESS : KFUPM P. BOX # 750
Dhahran 31261
Saudi Arabia

NATIONALITY : Saudi

EDUCATION

1992 - Date : King Fahd University of Petroleum and Minerals (KFUPM)
Mechanical Engineering Department, Dhahran, Saudi Arabia.

Expected to graduate by October 30, 1994 with a Master's Degree in Mechanical Engineering I finished Master's Courses with a GPA 4.00 on 4.00 scale. I took graduate courses related to power plants, advanced fluid mechanics, heat conduction, heat convection, numerical solution to PDEs, continuum mechanics and advanced engineering mathematics.

My M.S thesis was related to power plants. My thesis title is "Energy and Exergy Analysis of Ghazlan Power Plant". In this thesis the first and second law analysis were carried out using a computer program developed by me. Having a very good knowledge in numerical work which involve any finite difference or finite element techniques.

Very interested in researches related to energy applications, heat and fluid.

Having excellent knowledge in many programming languages like: FORTRAN and BASIC.

Familiar with P.C. usage and many technical P.C. programs.

Share in teaching some short courses like fluid power systems circuits and combustion.

1987 - 1992 King Fahd University of Petroleum and Minerals (KFUPM)

Mechanical Engineering Department, Dhahran, Saudi Arabia.

Graduate in January 1992 in Mechanical Engineering with a GPA 3.34 on 4.00 scale with Honors.

I took courses related to energy conversion, compressible fluid flow, fluid power systems, mechanical vibrations and system dynamics and control. familiar with machine design and design concepts. I designed a gear box and their design projects in machine design courses.

My Senior Project was related to gas and steam turbines.